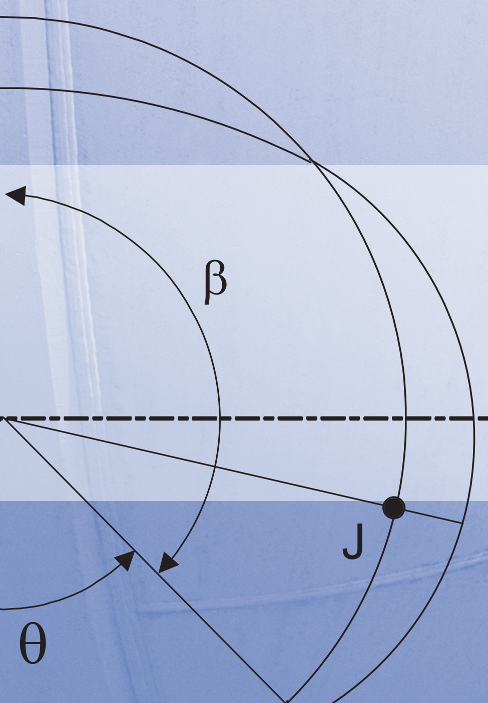
2010 ASME Boiler and Pressure Vessel Code

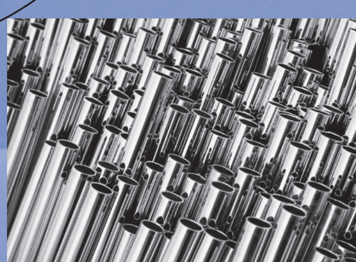
AN INTERNATIONAL CODE

VIII

Division 2
Alternative Rules

Rules for Construction of Pressure Vessels





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AN INTERNATIONAL CODE

2010 ASME Boiler & Pressure Vessel Code

2010 Edition July 1, 2010

VIII

Division 2

Alternative Rules

RULES FOR CONSTRUCTION OF PRESSURE VESSELS

ASME Boiler and Pressure Vessel Committee on Pressure Vessels

ASME SETTING THE STANDARD

Three Park Avenue • New York, NY • 10016 USA

Date of Issuance: July 1, 2010
(Includes all Addenda dated July 2009 and earlier)

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Library of Congress Catalog Card Number: 56-3934
Printed in the United States of America

Adopted by the Council of the American Society of Mechanical Engineers, 1914.
Revised 1940, 1941, 1943, 1946, 1949, 1952, 1953, 1956, 1959, 1962, 1965, 1968, 1971, 1974, 1977, 1980, 1983, 1986, 1989, 1992, 1995, 1998, 2001, 2004, 2007, 2010

The American Society of Mechanical Engineers
Three Park Avenue, New York, NY 10016-5990

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ADDENDA

Addenda, which include additions and revisions to individual Sections of the Code, will be sent automatically to purchasers of the applicable Sections up to the publication of the 2013 Code. The 2010 Code is available only in the loose-leaf format; accordingly, the Addenda will be issued in the loose-leaf, replacement-page format.

INTERPRETATIONS

ASME issues written replies to inquiries concerning interpretation of technical aspects of the Code. The Interpretations for each individual Section will be published separately and will be included as part of the update service to that Section. Interpretations of Section III, Divisions 1 and 2, will be included with the update service to Subsection NCA.

Interpretations of the Code are posted in January and July at www.cstools.asme.org/interpretations.

CODE CASES

The Boiler and Pressure Vessel Committee meets regularly to consider proposed additions and revisions to the Code and to formulate Cases to clarify the intent of existing requirements or provide, when the need is urgent, rules for materials or constructions not covered by existing Code rules. Those Cases that have been adopted will appear in the appropriate 2010 Code Cases book: "Boilers and Pressure Vessels" and "Nuclear Components." Supplements will be sent automatically to the purchasers of the Code Cases books up to the publication of the 2013 Code.

FOREWORD

(10)

The American Society of Mechanical Engineers set up a committee in 1911 for the purpose of formulating standard rules for the construction of steam boilers and other pressure vessels. This committee is now called the Boiler and Pressure Vessel Committee.

The Committee's function is to establish rules of safety, relating only to pressure integrity, governing the construction[1] of boilers, pressure vessels, transport tanks and nuclear components, and inservice inspection for pressure integrity of nuclear components and transport tanks, and to interpret these rules when questions arise regarding their intent. This code does not address other safety issues relating to the construction of boilers, pressure vessels, transport tanks and nuclear components, and the inservice inspection of nuclear components and transport tanks. The user of the Code should refer to other pertinent codes, standards, laws, regulations, or other relevant documents. With few exceptions, the rules do not, of practical necessity, reflect the likelihood and consequences of deterioration in service related to specific service fluids or external operating environments. Recognizing this, the Committee has approved a wide variety of construction rules in this Section to allow the user or his designee to select those which will provide a pressure vessel having a margin for deterioration in service so as to give a reasonably long, safe period of usefulness. Accordingly, it is not intended that this Section be used as a design handbook; rather, engineering judgment must be employed in the selection of those sets of Code rules suitable to any specific service or need.

This Code contains mandatory requirements, specific prohibitions, and nonmandatory guidance for construction activities. The Code does not address all aspects of these activities and those aspects which are not specifically addressed should not be considered prohibited. The Code is not a handbook and cannot replace education, experience, and the use of engineering judgment. The phrase *engineering judgment* refers to technical judgments made by knowledgeable designers experienced in the application of the Code. Engineering judgments must be consistent with Code philosophy and such judgments must never be used to overrule mandatory requirements or specific prohibitions of the Code.

The Committee recognizes that tools and techniques used for design and analysis change as technology progresses and expects engineers to use good judgment in the application of these tools. The designer is responsible for complying with Code rules and demonstrating compliance with Code equations when such equations are mandatory. The Code neither requires nor prohibits the use of computers for the design or analysis of components constructed to the requirements of the Code. However, designers and engineers using computer programs for design or analysis are cautioned that they are responsible for all technical assumptions inherent in the programs they use and they are responsible for the application of these programs to their design.

The Code does not fully address tolerances. When dimensions, sizes, or other parameters are not specified with tolerances, the values of these parameters are considered nominal and allowable tolerances or local variances may be considered acceptable when based on engineering judgment and standard practices as determined by the designer.

The Boiler and Pressure Vessel Committee deals with the care and inspection of boilers and pressure vessels in service only to the extent of providing suggested rules of good practice as an aid to owners and their inspectors.

The rules established by the Committee are not to be interpreted as approving, recommending, or endorsing any proprietary or specific design or as limiting in any way the manufacturer's freedom to choose any method of design or any form of construction that conforms to the Code rules.

The Boiler and Pressure Vessel Committee meets regularly to consider revisions of the rules, new rules as dictated by technological development, Code Cases, and requests for interpretations. Only the Boiler and Pressure Vessel Committee has the authority to provide official interpretations of this Code. Requests for revisions, new rules, Code Cases, or interpretations shall be addressed to the Secretary in writing and shall give full particulars in order to receive consideration and action (see Mandatory Appendix covering preparation of technical inquiries). Proposed revisions to the Code resulting from inquiries will be presented to the Main Committee for appropriate action. The action of the Main Committee becomes effective only after confirmation by letter ballot of the Committee and approval by ASME.

[1] *Construction*, as used in this Foreword, is an all-inclusive term comprising materials, design, fabrication, examination, inspection, testing, certification, and pressure relief.

Proposed revisions to the Code approved by the Committee are submitted to the American National Standards Institute and published at *http://cstools.asme.org/csconnect/public/index.cfm?PublicReview=Revisions* to invite comments from all interested persons. After the allotted time for public review and final approval by ASME, revisions are published in updates to the Code.

Code Cases may be used in the construction of components to be stamped with the ASME Code symbol beginning with the date of their approval by ASME.

After Code revisions are approved by ASME, they may be used beginning with the date of issuance. Revisions, except for revisions to material specifications in Section II, Parts A and B, become mandatory six months after such date of issuance, except for boilers or pressure vessels contracted for prior to the end of the six-month period. Revisions to material specifications are originated by the American Society for Testing and Materials (ASTM) and other recognized national or international organizations, and are usually adopted by ASME. However, those revisions may or may not have any effect on the suitability of material, produced to earlier editions of specifications, for use in ASME construction. ASME material specifications approved for use in each construction Code are listed in the Guidelines for Acceptable ASTM Editions and in the Guidelines for Acceptable Non-ASTM Editions, in Section II, Parts A and B. These Guidelines list, for each specification, the latest edition adopted by ASME, and earlier and later editions considered by ASME to be identical for ASME construction.

The Boiler and Pressure Vessel Committee in the formulation of its rules and in the establishment of maximum design and operating pressures considers materials, construction, methods of fabrication, inspection, and safety devices.

The Code Committee does not rule on whether a component shall or shall not be constructed to the provisions of the Code. The Scope of each Section has been established to identify the components and parameters considered by the Committee in formulating the Code rules.

Questions or issues regarding compliance of a specific component with the Code rules are to be directed to the ASME Certificate Holder (Manufacturer). Inquiries concerning the interpretation of the Code are to be directed to the ASME Boiler and Pressure Vessel Committee. ASME is to be notified should questions arise concerning improper use of an ASME Code symbol.

The specifications for materials given in Section II are identical with or similar to those of specifications published by ASTM, AWS, and other recognized national or international organizations. When reference is made in an ASME material specification to a non-ASME specification for which a companion ASME specification exists, the reference shall be interpreted as applying to the ASME material specification. Not all materials included in the material specifications in Section II have been adopted for Code use. Usage is limited to those materials and grades adopted by at least one of the other Sections of the Code for application under rules of that Section. All materials allowed by these various Sections and used for construction within the scope of their rules shall be furnished in accordance with material specifications contained in Section II or referenced in the Guidelines for Acceptable Editions in Section II, Parts A and B, except where otherwise provided in Code Cases or in the applicable Section of the Code. Materials covered by these specifications are acceptable for use in items covered by the Code Sections only to the degree indicated in the applicable Section. Materials for Code use should preferably be ordered, produced, and documented on this basis; Guidelines for Acceptable Editions in Section II, Part A and Guidelines for Acceptable Editions in Section II, Part B list editions of ASME and year dates of specifications that meet ASME requirements and which may be used in Code construction. Material produced to an acceptable specification with requirements different from the requirements of the corresponding specifications listed in the Guidelines for Acceptable Editions in Part A or Part B may also be used in accordance with the above, provided the material manufacturer or vessel manufacturer certifies with evidence acceptable to the Authorized Inspector that the corresponding requirements of specifications listed in the Guidelines for Acceptable Editions in Part A or Part B have been met. Material produced to an acceptable material specification is not limited as to country of origin.

When required by context in this Section, the singular shall be interpreted as the plural, and vice-versa; and the feminine, masculine, or neuter gender shall be treated as such other gender as appropriate.

STATEMENT OF POLICY ON THE USE OF CODE SYMBOLS AND CODE AUTHORIZATION IN ADVERTISING

(10)

ASME has established procedures to authorize qualified organizations to perform various activities in accordance with the requirements of the ASME Boiler and Pressure Vessel Code. It is the aim of the Society to provide recognition of organizations so authorized. An organization holding authorization to perform various activities in accordance with the requirements of the Code may state this capability in its advertising literature.

Organizations that are authorized to use Code Symbols for marking items or constructions that have been constructed and inspected in compliance with the ASME Boiler and Pressure Vessel Code are issued Certificates of Authorization. It is the aim of the Society to maintain the standing of the Code Symbols for the benefit of the users, the enforcement jurisdictions, and the holders of the symbols who comply with all requirements.

Based on these objectives, the following policy has been established on the usage in advertising of facsimiles of the symbols, Certificates of Authorization, and reference to Code construction. The American Society of Mechanical Engineers does not "approve," "certify," "rate," or "endorse" any item, construction, or activity and there shall be no statements or implications that might so indicate. An organization holding a Code Symbol and/or a Certificate of Authorization may state in advertising literature that items, constructions, or activities "are built (produced or performed) or activities conducted in accordance with the requirements of the ASME Boiler and Pressure Vessel Code," or "meet the requirements of the ASME Boiler and Pressure Vessel Code." An ASME corporate logo shall not be used by any organization other than ASME.

The ASME Symbol shall be used only for stamping and nameplates as specifically provided in the Code. However, facsimiles may be used for the purpose of fostering the use of such construction. Such usage may be by an association or a society, or by a holder of a Code Symbol who may also use the facsimile in advertising to show that clearly specified items will carry the symbol. General usage is permitted only when all of a manufacturer's items are constructed under the rules.

STATEMENT OF POLICY ON THE USE OF ASME MARKING TO IDENTIFY MANUFACTURED ITEMS

The ASME Boiler and Pressure Vessel Code provides rules for the construction of boilers, pressure vessels, and nuclear components. This includes requirements for materials, design, fabrication, examination, inspection, and stamping. Items constructed in accordance with all of the applicable rules of the Code are identified with the official Code Symbol Stamp described in the governing Section of the Code.

Markings such as "ASME," "ASME Standard," or any other marking including "ASME" or the various Code Symbols shall not be used on any item that is not constructed in accordance with all of the applicable requirements of the Code.

Items shall not be described on ASME Data Report Forms nor on similar forms referring to ASME that tend to imply that all Code requirements have been met when, in fact, they have not been. Data Report Forms covering items not fully complying with ASME requirements should not refer to ASME or they should clearly identify all exceptions to the ASME requirements.

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As of January 1, 2010

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SUMMARY OF CHANGES

Record Numbers listed below are explained in more detail in "List of Changes in Record Number Order" following the Summary of Changes.

Page	Location	Change (Record Number)
iii	Table Of Contents	Update for all changes (10-114)
1-8	Table 1.1	Correct revision dates in Table 1.1 and correct numbering on rollover of Table 1.1 to next page (07-1873)
2-8	Paragraph 2.4.2	Delete phrase "sovereign government " and replace with jurisdiction (08-108)
2-17	Paragraph 2.C.3.1.d.1	Provide elaboration on repair records (07-251)
2-23	Table 2.D.3 Form A-1	Item 5 Replace "data" with date (08-361)
2-23	Table 2.D.3 Form A-1	Item 12 replace "wid" with "wld" (08-361)
2-23	Table 2.D.3 Form A-1	Correct description item15 Notes 22 and 24 to "Nondestructive Examination" (09-1918)
2-23	Table 2.D.3 Form A-2	Correct description item 8 and 16 Notes 22 and 24 to "Nondestructive Examination" (09-1918)
2-24	Table 2.D.3 Form A-1	Item 45 line 7 delete psi (08-361)
2-25	Table 2.D.3 Form A-2	Item 4 delete one period (08-361)
2-25	Table 2.D.3 Form A-2	Item 5 Replace "data" with date (08-361)
2-25	Table 2.D.3 Form A-2	Item 12 replace "wid" with "wld" (08-361)
2-26	Table 2.D.3 Form A-2	Item 43 Lines 5 and 6 replace "pressure vessel" with "part" (08-361)
2-26	Table 2.D.3 Form A-2	Item 43 Line 4 Add underline after word "on" (08-361)
2-42	Table 2.H.1.Note 2	Add 3 additional example scope statements for UV Code Symbol Stamp (09-263)
3-6	Paragraph 3.2.6.1	Correct paragraph header (09-1782)
3-10	Paragraph 3.3.6.2	Delete "and" replace with "or" and add text to end of paragraph prior to list (09-376)
3-13	Paragraph 3.6.3.3.b & c	Replace word "approximate" with "appropriate"(09-1800)
3-24	Paragraph 3.11.2.2.a	Revise temperature from 32°C (90°F) to 43°C (110°F) (09-313)
3-30	Paragraph 3.11.4.3.a.2	Correct subparagraph reference (09-1335)
3-30	Paragraph 3.11.4.3.a.3	Delete word "not" line1 after word "content" (09-1335)
3-81	Table 3.A.3	Add SA-240 S32550 (09-204)
3-85	Table 3.A.3	Add entry for SA-564, SA-693, and SA-705 (03-1422)
3-88	Table 3.A.6	Change Table Title to Nickel and Nickel Alloys (08-765)
3-101	Paragraph 3.D.3	Provide additional information (09-1800)
3-101	Paragraph 3.D.3	Add Equation 3.D.13, renumber all equations(09-1800)
3-102	Paragraph 3.D.5.1	Add text pointing to equation for K (09-1800)
3-103	Nomenclature	Add definition for K and $\sigma_{uts,t}$ (09-1800)
3-105	Table 3.D.2	Replace C_{css} with K_{css} (09-1800)
3-110	Equation 3.F.1	Correct equation (09-1800)
3-110	Equation 3.F.2	Correct equation (09-1800)
3-110	Equation 3.F.3	Add equation and renumber all other equations and references(09-1800)
3-111	Equation 3.F.8	Correct constant in denominator(09-1800)
3-112	Nomenclature	Add definition for C_{usm} and Y (09-1800)
3-124	Table 3.F.11	Replace ΔS_{range} with $\Delta S_{ess,k}$ (09-1800)

Page	Location	Change (Record Number)
3-124	Table 3.F.11M	Replace ΔS_{range} with $\Delta S_{ess,k}$ (09-1800)
4-7	Paragraph 4.1.2.d.3	Revise to provide lower limit (09-375)
4-10	Paragraph 4.1.8.2	Revise paragraph for clarity (08-1560)
4-13	Table 4.1.1	Amend description for Load Parameter L (08-1499)
4-17	Paragraph 4.2.5.5.c.4	Add word “continuous” before word “weld” in line 2 (09-535)
4-34	Table 4.2.10	Corrections to design notes for r_1 *and* r_3 (09-18)
4-37	Table 4.2.11	Corrections to design notes for r_1 *and* r_3 (09-18)
4-39	Table 4.2.12	Corrections to design notes for r_1 *and* r_3 (09-18)
4-40	Table 4.2.13	Corrections to design notes for r_1 (09-18)
4-56	Paragraph 4.3.11.4.f	Replace “average circumferential” with “hoop” and “average longitudinal” with “axial” (08-613)
4-56	Paragraph 4.3.11.4.f	Replace $\sigma_{\theta am}$ with $\sigma_{\theta m}$ and σ_{sam} with σ_{sm} (08-613)
4-56	Paragraph 4.3.11.4.f	Revise text (08-613)
4-56	Equations 4.3.54 and 4.3.55	Correct variables (08-613)
4-57	Paragraph 4.3.11.5.f	Replace “average circumferential” with “hoop” and “average longitudinal” with “axial” (08-613)
4-57	Paragraph 4.3.11.5.f	Replace $\sigma_{\theta am}$ with $\sigma_{\theta m}$ and σ_{sam} with σ_{sm} (08-613)
4-57	Paragraph 4.3.11.5.f	Revise text (08-613)
4-58	Paragraph 4.3.12.2.e	Replace “average circumferential” with “hoop” and “average longitudinal” with “axial” (08-613)
4-58	Paragraph 4.3.12.2.e	Replace $\sigma_{\theta am}$ with $\sigma_{\theta m}$ and σ_{sam} with σ_{sm} (08-613)
4-58	Paragraph 4.3.12.2.e	Revise text (08-613)
4-58	Equations 4.3.67 and 4.3.68	Correct variables (08-613)
4-59	Paragraph 4.3.12.3.e	Replace “average circumferential” with “hoop” and “average longitudinal” with “axial” (08-613)
4-59	Paragraph 4.3.12.3.e	Replace $\sigma_{\theta am}$ with $\sigma_{\theta m}$ and σ_{sam} with σ_{sm} (08-613)
4-59	Paragraph 4.3.12.3.e	Revise text (08-613)
4-61	Nomenclature	Delete definition for $\sigma_{\theta am}$ and σ_{sam} (08-613)
4-61	Nomenclature	Add definition for P_a, P_{eth} Change definition for P_{ac}, P_{ak} delete extra P_{ac}, delete P_{ah} (09-1800)
4-62	Nomenclature	Replace “circumferential” with “hoop (circumferential)” for σ_θ, $\sigma_{\theta m}$, $\sigma_{\theta b}$ and “meridional” with “axial (longitudinal)” for σ_s, σ_{sm}, σ_{sb} (08-613)
4--68	Table 4.3.5	Correct values for equation coefficients (09-1800)
4-69	Table 4.3.6	Correct values for equation coefficients (09-1800)
4-70	Table 4.3.7	See RC 09-1800 (09-11-09)
4-70	Table 4.3.7	Delete - in rows 1 and 4 replace with :(09-1800)
4-71	Table 4.3.7	Correct equations for stress in non compact knuckle region (09-1800)
4-72	Table 4.3.8	See RC 09-1800 (09-11-09)
4-72	Table 4.3.8	Delete - in rows 1 and 4 replace with :(09-1800)

Page	Location	Change (Record Number)
4-73	Table 4.3.8	Correct equations for stress in non compact flare region (09-1800)
4-89	Equation 4.4.55	Correct equation ratio in denominator was inverted (08-613)
4-93	Paragraph 4.4.12.2.d	Replace word “shall” with “shell”(09-1782)
4-96	Paragraph 4.4.12.2.i.3	Correct paragraph references (09-1782)
4-96	Paragraph 4.4.12.2.i.3.	Correct paragraph references (09-1800)
4-99	Paragraph 4.4.13.2	Revise paragraph (08-613)
4-99	Paragraph 4.4.14	Add subparagraphs 4.4.14.1 and 4.4.14.2 (08-613)
4-102	Nomenclature	Correct definition for R , R_L , R_S , Z_c add definition for R_m (09-1800)
4-104	Table 4.4.1	Add reference to Part 3.A Table 3.A.1 (08-765)
4-107	Figure 4.4.3a	Correct figure (09-1800)
4-113	Paragraph 4.5.4.1	Add sentence on corrosion allowance (08-631)
4-134	Table 4.5.2	Replace existing table with new table (08-631)
4-143	Equation 4.6.6	Correct equation (09-714)
4-144	Paragraph 4.6.4.3.d	Add reference to Table 4.16.5 (09-714)
4-144	Paragraph 4.6.4.3.e	Add reference to Table 4.16.5 (09-714)
4-144	Paragraph 4.6.4.3.f	Correct reference to Table 4.16.4 (09-714)
4-145	Nomenclature	Correct definition for h_G (09-1800)
4-180	Equation 4.11.4	Correct equation (09-1800)
4-206	Nomenclature	Correct definition for Δ (09-1800)
4-267	Nomenclature	Corrections to definitions for r_1 and r_3 (09-18)
4-286	Paragraph 4.15.3.3.c.1	Delete E replace with SE . (07-1352)
4-286	Paragraph 4.15.3.3.c.2	Revise paragraph references(07-1352)
4-293	Nomenclature	Add definition for E (07-1352)
4-307	Paragraph 4.16.6.1.c.2	Correct item numbering (09-1800)
4-309	Paragraph 4.16.7.2.f	Insert statements concerning maximum bolt spacing and flange moments in the calculation of rigidity index (07-909)
4-309	Equation 4.16.14	Correct equation to include bolt spacing correction (07-909)
4-309	Equation 4.16.17	Correct equation to include bolt spacing correction (07-909)
4-310	Paragraph 4.16.7.2.j	Insert statements concerning flange moments in the calculation of rigidity index (07-909)
4-311	Nomenclature	Add definition for $a, B_s, B_{s\max}, B_{sc}$ (07-909)
4-326	Table 4.16.10	Insert Reverse integral type flange, reverse loose type flange with a hub, and add reverse loose type to loose type flange without hub (07-909)
4-326	Table 4.16.11	Add table for bolt spacing equations (07-909)
4-341	Nomenclature	Delete extra definition for N (09-1800)
4-345	Figure 4.17.1	Correct detail d and Note 1(09-1800)
4-356	Paragraph 4.18.7.5.3	Correct $S_{PS,c}$ and $S_{PS,s}$ (10-114)
4-357	Paragraph 4.18.8.3.a	Insert phrase “tube-to-tubesheet joint” (05-484)
4-358	Paragraph 4.18.8.3.c	Insert phrase “and that the resulting tube-to-tubesheet joint load is acceptable” (05-484)
4-358	Paragraph 4.18.8.3.d	Delete word “or” between shell and channel Insert phrase “or tube-to-tubesheet joint,” (05-484)
4-358	Paragraph 4.18.8.3.e.2	Delete item 2 (05-484)
4-363	Paragraph 4.18.8.4.i	STEP 9 insert phrase “and tube-to-tubesheet joint design “ after word stress (05-484)

Page	Location	Change (Record Number)
4-364	Paragraph 4.18.8.4.i.2-6	Revise section(05-484)
4-364	Paragraph 4.18.8.4.l.2 & 3	STEP 12 Expand option 2 and delete subparagraphs in option 3 (06-17)
4-365	Paragraph 4.18.8.4.i.6.iv	Add word “If” to beginning of sentence. (10-114)
4-366	Paragraph 4.18.8.4.k.2	Add comma after word “cylindrical” in line 1 (10-114)
4-367	Paragraph 4.18.8.4.l.2-3	Revise section (06-17)
4-368	Paragraph 4.18.8.5.c.5	Correct $S_{sb,1}$ with $S_{s,b,1}$ (10-114)
4-368	Paragraph 4.18.8.6.2	Replace “Young’s modulus” with “modulus of elasticity” (06-17)
4-371	Paragraph 4.18.8.8.2	Correct spelling for “integral” (10-114)
4-372	Paragraph 4.18.8.8.3.b.3.i, ii	Correct $S_{PS,c}$ and $S_{PS,s}$ (10-114)
4-372	Paragraph 4.18.8.8.3.d.1 , 2	Correct numbering and change item 2 to configuration a only(10-114)
4-373	Paragraph 4.18.9.3.b	Insert phrase “tube-to-tubesheet joint” (05-484)
4-374	Paragraph 4.18.9.3.d	Insert phrase “and that the resulting tube-to-tubesheet joint load is acceptable” (05-484)
4-374	Paragraph 4.18.9.3.e.2	Delete item 2 and renumber (05-484)
4-378	Paragraph 4.18.9.4.i	STEP 9 insert phrase “and tube-to-tubesheet joint design “ after word stress (05-484)
4-379	Paragraph 4.18.9.4.i.2-6	Revise section(05-484)
4-381	Paragraph 4.18.9.4.k	STEP 11 add option 3 ((06-17)
4-382	Paragraph 4.18.9.5 new	Add calculation procedure for effect of plasticity at tubesheet/channel or shell joint for floating tubesheets (06-17)
4-383	Paragraph 4.18.9.5	Move Paragraph 4.18.9.5 to 4.18.9.7
4-385	Paragraph 4.18.9.7.2	Correct spelling for “integral” (10-114)
4-385	Paragraph 4.18.9.7.3.b.3.i, ii	Correct $S_{PS,c}$ and $S_{PS,s}$ (10-114)
4-385	Paragraph 4.18.9.7.3.d.1 , 2	Correct numbering (10-114)
4-391	Paragraph 4.18.14.3	Correct reference to paragraph 4.1.8.2 and revise for clarity. (08-1560)
4-394	Nomenclature	Correct $S_{s,b,1}$ (10-114)
4-395	Nomenclature	Add definition for W_t (05-484)
4-401	Table 4.18.4	Various corrections (10-114)
4-417	Paragraph 4.19.2.e	Revise paragraph (08-830)
4-417	Paragraph 4.19.2.f	Add paragraph f (08-830)
4-418	Paragraph 4.19.3.1.h	Revise paragraph
4-457	Paragraph 4.C.1.1	Delete word welds (01-576)
4-457	Paragraph 4.C.1.1.a	Insert phrase” tube-to-tubesheet joints having”
4-457	Paragraph 4.C.1.1.b	Insert phrase “tube-to-tubesheet joints having” and word “as” delete phrase “shall be” and word “and” (01-576)
4-457	Paragraph 4.C.1.3.b	Delete word “provided” insert word “if”
4-457	Paragraph 4.C.1.5.b.1	Rephrase item (01-576)
4-458	Paragraph 4.C.1.5.b.2	Insert reference to 4.C.1.5.b.4 (01-576)
4-458	Paragraph 4.C.1.5.b.4	Insert reference to 4.C.1.5.b.2 (01-576)
4-458	Paragraph 4.C.2	Rephrase paragraph (01-576)
4-459	Paragraph 4.C.3.3	Insert space between E and 8 (01-576)
4-460	Paragraph 4.C.3.11.a	Insert word “the” in line 3 (01-576)
4-460	Paragraph 4.C.4.2	Rephrase and insert symbols (01-576)
4-460	Paragraph 4.C.5	Delete word “met” and replace with “satisfied” (01-576)

Page	Location	Change (Record Number)
4-461	Nomenclature	Correct definition for f_e, k, P_o (01-576)
4-461	Nomenclature	Add definition for S_y, S_{ua} (01-576)
5-16	Equation 5.19	Capitalize Variable “p” (09-1782)
5-16	Equation 5.19	Capitalize Variable “p” (09-1800)
5-20	Paragraph 5.5.3.2.e	Add paragraph 3.F.1 (09-1800)
5-22	Paragraph 5.5.4.2.g	Add paragraph 3.F.1 (09-1800)
5-25	Paragraph 5.5.5.2.g	Add paragraph 3.F.2 (09-1800)
5-43	Table 5.3	Add note (09-75)
5-44	Table 5.4	Add note and correct equations for global criteria (09-75)
5-45	Table 5.5	Add note and correct equations for global criteria (09-75)
5-51	Table 5.11	Correct values (09-1800)
5-61	Nomenclature	Add definition for σ_s (09-1800)
5-145	Figure 5.F.1	Remove phrase “For Accelerated Tests” from title (09-1604)
5-146	Figure 5.F.2	Add phrase “For Accelerated Tests” to title (09-1604)
6-19	Paragraph 6.4.3.8	Add “such as “spot” or “bulls eye” local heating” after word configurations in line 1 (09-1832)
6-22	Paragraph 6.5.5.1	Delete NOTE after 6.5.5.1.b (07-347)
6-38	Table 6.1	Delete last row in table (05-151)
6-66	Figure 6.6	Delete figure (05-151)
7-15	Paragraph 7.5.5.1	Delete paragraphs b, c, d, and f, renumber and edit remaining paragraphs (09-1918)
7-16	Paragraph 7.5.5.2,	Delete paragraphs and renumber (09-1918)
7-16	Paragraph 7.5.5.3	Delete paragraphs and renumber (09-1918)
7-31	Table 7.8	Correct paragraph references (09-1918)
7-48	Figures 7.16 and 7.17	Add figures (09-1943)
8-2	Paragraph 8.1.2.d	Expand safety requirements for air or gas pressure testing (07-256)
8-3	Paragraph 8.1.3.2	Delete paragraph 8.1.3.2.d (08-75)
8-3	8.1.3.4	Delete paragraph 8.1.3.4 (08-75)

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LIST OF CHANGES IN RECORD NUMBER ORDER

RC Number	Location	Change (Record Number)
01-576	Paragraph 4.C.1.1	Delete word welds (01-576)
	Paragraph 4.C.1.1.a	Insert phrase" tube-to-tubesheet joints having"
	Paragraph 4.C.1.1.b	Insert phrase "tube-to-tubesheet joints having" and word "as" delete phrase "shall be" and word "and" (01-576)
	Paragraph 4.C.1.3.b	Delete word "provided" insert word "if"
	Paragraph 4.C.1.5.b.1	Rephrase item (01-576)
	Paragraph 4.C.1.5.b.2	Insert reference to 4.C.1.5.b.4 (01-576)
	Paragraph 4.C.1.5.b.4	Insert reference to 4.C.1.5.b.2 (01-576)
	Paragraph 4.C.2	Rephrase paragraph (01-576)
	Paragraph 4.C.3.3	Insert space between E and 8 (01-576)
	Paragraph 4.C.3.11.a	Insert word "the" in line 3 (01-576)
	Paragraph 4.C.4.2	Rephrase and insert symbols (01-576)
	Paragraph 4.C.5	Delete word "met" and replace with "satisfied" (01-576)
	Nomenclature	Correct definition for f_e, k, P_o (01-576)
	Nomenclature	Add definition for S_y, S_{ua} (01-576)
03-1422	Table 3.A.3	Add entry for SA-564, SA-693, and SA-705 (03-1422)
05-151	Table 6.1	Delete last row in table (05-151)
	Figure 6.6	Delete figure (05-151)
05-484	Paragraph 4.18.8.3.a	Insert phrase "tube-to-tubesheet joint" (05-484)
	Paragraph 4.18.8.3.c	Insert phrase "and that the resulting tube-to-tubesheet joint load is acceptable" (05-484)
	Paragraph 4.18.8.3.d	Delete word "or" between shell and channel Insert phrase "or tube-to-tubesheet joint," (05-484)
	Paragraph 4.18.8.3.e.2	Delete item 2 (05-484)
	Paragraph 4.18.8.4.i	STEP 9 insert phrase "and tube-to-tubesheet joint design " after word stress (05-484)
	Paragraph 4.18.8.4.i.2-6	Revise section(05-484)
	Paragraph 4.18.9.3.b	Insert phrase "tube-to-tubesheet joint" (05-484)
	Paragraph 4.18.9.3.d	Insert phrase "and that the resulting tube-to-tubesheet joint load is acceptable" (05-484)
	Paragraph 4.18.9.3.e.2	Delete item 2 and renumber (05-484)
	Paragraph 4.18.9.4.i	STEP 9 insert phrase "and tube-to-tubesheet joint design " after word stress (05-484)
	Paragraph 4.18.9.4.i.2-6	Revise section(05-484)
	Nomenclature	Add definition for W_t (05-484)
06-17	Paragraph 4.18.8.4.l.2 & 3	STEP 12 Expand option 2 and delete subparagraphs in option 3 (06-17)
	Paragraph 4.18.9.4.k	STEP 11 add option 3 (06-17)
	Paragraph 4.18.8.6	Replace "Young's modulus" with "modulus of elasticity" (06-17)
	Paragraph 4.18.9.5 new	Add calculation procedure for effect of plasticity at tubesheet/channel or shell joint for floating tubesheets (06-17)
07-1352	Paragraph 4.15.3.3.c.1	Delete E replace with SE. (07-1352)
	Paragraph 4.15.3.3.c.2	Revise paragraph references(07-1352)
	Nomenclature	Add definition for E (07-1352)

RC Number	Location	Change (Record Number)
07-1873	Table 1.1	Correct revision dates in Table 1.1 and correct numbering on rollover of Table 1.1 to next page (07-1873)
07-251	Paragraph 2.C.3.1.d.1	Provide elaboration on repair records (07-251)
07-256	Paragraph 8.1.2.d	Expand safety requirements for air or gas pressure testing (07-256)
07-347	Paragraph 6.5.5.1	Delete NOTE after 6.5.5.1.b (07-347)
07-909	Paragraph 4.16.7.2.f	Insert statements concerning maximum bolt spacing and flange moments in the calculation of rigidity index (07-909)
	Equation 4.16.14	Correct equation to include bolt spacing correction (07-909)
	Equation 4.16.17	Correct equation to include bolt spacing correction (07-909)
	Paragraph 4.16.7.2.j	Insert statements concerning flange moments in the calculation of rigidity index (07-909)
	Nomenclature	Add definition for $a, B_s, B_{s\max}, B_{sc}$ (07-909)
	Table 4.16.10	Insert Reverse integral type flange, reverse loose type flange with a hub, and add reverse loose type to loose type flange without hub (07-909)
	Table 4.16.11	Add table for bolt spacing equations (07-909)
08-108	Paragraph 2.4.2	Delete phrase “sovereign government “ and replace with jurisdiction (08-108)
08-1499	Table 4.1.1	Amend description for Load Parameter L (08-1499)
08-1560	Paragraph 4.1.8.2	Revise paragraph for clarity (08-1560)
	Paragraph 4.18.14.3	Correct reference to paragraph 4.1.8.2 and revise for clarity. (08-1560)
08-361	Table 2.D.3 Form A-1	Item 5 Replace “data” with date (08-361)
	Table 2.D.3 Form A-1	Item 12 replace “wid” with “wld” (08-361)
	Table 2.D.3 Form A-1	Item 45 line 7 delete psi (08-361)
	Table 2.D.3 Form A-2	Item 4 delete one period (08-361)
	Table 2.D.3 Form A-2	Item 5 Replace “data” with date (08-361)
	Table 2.D.3 Form A-2	Item 12 replace “wid” with “wld” (08-361)
	Table 2.D.3 Form A-2	Item 43 Lines 5 and 6 replace “pressure vessel” with “part” (08-361)
	Table 2.D.3 Form A-2	Item 43 Line 4 Add underline after word “on” (08-361)
08-613	Paragraph 4.3.11.4.f	Replace “average circumferential” with “hoop” and “average longitudinal” with “axial” (08-613)
	Paragraph 4.3.11.4.f	Replace $\sigma_{\theta am}$ with $\sigma_{\theta m}$ and σ_{sam} with σ_{sm} (08-613)
	Paragraph 4.3.11.4.f	Revise text (08-613)
	Equations 4.3.54 and 4.3.55	Correct variables (08-613)
	Paragraph 4.3.11.5.f	Replace “average circumferential” with “hoop” and “average longitudinal” with “axial” (08-613)
	Paragraph 4.3.11.5.f	Replace $\sigma_{\theta am}$ with $\sigma_{\theta m}$ and σ_{sam} with σ_{sm} (08-613)
	Paragraph 4.3.11.5.f	Revise text (08-613)
	Paragraph 4.3.12.2.e	Replace “average circumferential” with “hoop” and “average longitudinal” with “axial” (08-613)
	Paragraph 4.3.12.2.e	Replace $\sigma_{\theta am}$ with $\sigma_{\theta m}$ and σ_{sam} with σ_{sm} (08-613)
	Paragraph 4.3.12.2.e	Revise text (08-613)
	Equations 4.3.67 and 4.3.68	Correct variables (08-613)

RC Number	Location	Change (Record Number)
08-613	Paragraph 4.3.12.3.e	Replace “average circumferential” with “hoop” and “average longitudinal” with “axial” (08-613)
	Paragraph 4.3.12.3.e	Replace $\sigma_{\theta am}$ with $\sigma_{\theta m}$ and σ_{sam} with σ_{sm} (08-613)
	Paragraph 4.3.12.3.e	Revise text (08-613)
	Nomenclature	Delete definition for $\sigma_{\theta am}$ and σ_{sam} (08-613)
	Nomenclature	Replace “circumferential” with “hoop (circumferential)” for σ_{θ}, $\sigma_{\theta m}$, $\sigma_{\theta b}$ and “meridional” with “axial (longitudinal)” for σ_{s}, σ_{sm}, σ_{sb} (08-613)
	Equation 4.4.55	Correct equation ratio in denominator was inverted (08-613)
	Paragraph 4.4.13.2	Revise paragraph (08-613)
	Paragraph 4.4.14	Add subparagraphs 4.4.14.1 and 4.4.14.2 (08-613)
08-631	Paragraph 4.5.4.1	Add sentence on corrosion allowance (08-631)
	Table 4.5.2	Replace existing table with new table (08-631)
08-75	Paragraph 8.1.3.2	Delete paragraph 8.1.3.2.d (08-75)
	8.1.3.4	Delete paragraph 8.1.3.4 (08-75)
08-765	Table 3.A.6	Change Table Title to Nickel and Nickel Alloys (08-765)
	Table 4.4.1	Add reference to Part 3.A Table 3.A.1 (08-765)
08-830	Paragraph 4.19.2.e	Revise paragraph (08-830)
	Paragraph 4.19.2.f	Add paragraph f (08-830)
	Paragraph 4.19.3.1.h	Revise paragraph
09-11-09	Table 4.3.7	See RC 09-1800 (09-11-09)
	Table 4.3.8	See RC 09-1800 (09-11-09)
09-1335	Paragraph 3.11.4.3.a.2	Correct subparagraph reference (09-1335)
	Paragraph 3.11.4.3.a.3	Delete word “not” line1 after word “content” (09-1335)
09-1604	Figure 5.F.1	Remove phrase “For Accelerated Tests” from title (09-1604)
	Figure 5.F.2	Add phrase “For Accelerated Tests” to title (09-1604)
09-1782	Paragraph 3.2.6.1	Correct paragraph header (09-1782)
	Paragraph 4.4.12.2.d	Replace word “shall” with “shell”(09-1782)
	4.4.12.2.i.3	Correct paragraph references (09-1782)
	Equation 5.19	Capitalize Variable “p” (09-1782)
09-18	Table 4.2.10	Corrections to design notes for r_1 *and* r_3(09-18)
	Table 4.2.11	Corrections to design notes for r_1 *and* r_3 (09-18)
	Table 4.2.12	Corrections to design notes for r_1 *and* r_3 (09-18)
	Table 4.2.13	Corrections to design notes for r_1 (09-18)
	Nomenclature	Corrections to definitions for r_1 and r_3 (09-18)
09-1800	Paragraph 3.6.3.3.b & c	Replace word “approximate” with “appropriate”(09-1800)
	Paragraph 3.D.3	Provide additional information (09-1800)
	Paragraph 3.D.3	Add Equation 3.D.13, renumber all equations(09-1800)
	Paragraph 3.D.5.1	Add text pointing to equation for K (09-1800)
	Nomenclature	Add definition for K and $\sigma_{uts,t}$ (09-1800)
	Table 3.D.2	Replace C_{css} with K_{css} (09-1800)
	Equation 3.F.1	Correct equation (09-1800)
	Equation 3.F.2	Correct equation (09-1800)

RC Number	Location	Change (Record Number)
	Equation 3.F.3	Add equation and renumber all other equations and references(09-1800)
09-1800	Equation 3.F.8	Correct constant in denominator(09-1800)
	Nomenclature	Add definition for C_{usm} and Y (09-1800)
	Table 3.F.11	Replace ΔS_{range} with $\Delta S_{ess,k}$ (09-1800)
	Table 3.F.11M	Replace ΔS_{range} with $\Delta S_{ess,k}$ (09-1800)
	Nomenclature	Add definition for P_a, P_{eth} Change definition for P_{ac}, P_{ak} delete extra P_{ac}, delete P_{ah} (09-1800)
	Table 4.3.5	Correct values for equation coefficients (09-1800)
	Table 4.3.6	Correct values for equation coefficients (09-1800)
	Table 4.3.7	Delete - in rows 1 and 4 replace with :(09-1800)
	Table 4.3.7	Correct equations for stress in non compact knuckle region (09-1800)
	Table 4.3.8	Delete - in rows 1 and 4 replace with :(09-1800)
	Table 4.3.8	Correct equations for stress in non compact flare region (09-1800)
	Paragraph 4.4.12.2.i.3.	Correct paragraph references (09-1800)
	Nomenclature	Correct definition for R, R_L, R_S, Z_c add definition for R_m (09-1800)
	Figure 4.4.3a	Correct figure (09-1800)
	Nomenclature	Correct definition for h_G (09-1800)
	Equation 4.11.4	Correct equation (09-1800)
	Nomenclature	Correct definition for Δ (09-1800)
	Paragraph 4.16.6.1.c.2	Correct item numbering (09-1800)
	Nomenclature	Delete extra definition for N (09-1800)
	Figure 4.17.1	Correct detail d and Note 1(09-1800)
	Equation 5.19	Capitalize Variable “p” (09-1800)
	Paragraph 5.5.3.2.e	Add paragraph 3.F.1 (09-1800)
	Paragraph 5.5.4.2.g	Add paragraph 3.F.1 (09-1800)
	Paragraph 5.5.5.2.g	Add paragraph 3.F.2 (09-1800)
	Table 5.11	Correct values (09-1800)
	Nomenclature	Add definition for σ_s (09-1800)
09-1832	Paragraph 6.4.3.8	Add “such as “spot” or “bulls eye” local heating” after word configurations in line 1 (09-1832)
09-1918	Table 2.D.3 Form A-1	Correct description item15 Notes 22 and 24 to “Nondestructive Examination” (09-1918)
	Table 2.D.3 Form A-2	Correct description item 8 and 16 Notes 22 and 24 to “Nondestructive Examination” (09-1918)
	Paragraph 7.5.5.1	Delete paragraphs b, c, d, and f, renumber and edit remaining paragraphs (09-1918)
	Paragraph 7.5.5.2,	Delete paragraphs and renumber (09-1918)
	Paragraph 7.5.5.3	Delete paragraphs and renumber (09-1918)
	Table 7.8	Correct paragraph references (09-1918)
09-1943	Figures 7.16 and 7.17	Add figures (09-1943)
09-204	Table 3.A.3	Add SA-240 S32550 (09-204)

RC Number	Location	Change (Record Number)
09-263	Table 2.H.1.Note 2	Add 3 additional example scope statements for UV Code Symbol Stamp (09-263)
09-313	Paragraph 3.11.2.2.a	Revise temperature from 32°C (90°F) to 43°C (110°F) (09-313)
09-375	Paragraph 4.1.2.d.3	Revise to provide lower limit (09-375)
09-376	Paragraph 3.3.6.2	Delete "and" replace with "or" and add text to end of paragraph prior to list (09-376)
09-535	Paragraph 4.2.5.5.c.4	Add word "continuous" before word "weld" in line 2 (09-535)
09-714	Equation 4.6.6	Correct equation (09-714)
	Paragraph 4.6.4.3.d	Add reference to Table 4.16.5 (09-714)
	Paragraph 4.6.4.3.e	Add reference to Table 4.16.5 (09-714)
	Paragraph 4.6.4.3.f	Correct reference to Table 4.16.4 (09-714)
09-75	Table 5.3	Add note (09-75)
	Table 5.4	Add note and correct equations for global criteria (09-75)
	Table 5.5	Add note and correct equations for global criteria (09-75)
10-114	Table Of Contents	Update for all changes (10-114)
	Paragraph 4.18.7.5.3	Correct $S_{PS,c}$ and $S_{PS,s}$ (10-114)
	Paragraph 4.18.8.4.i.6.iv	Add word "If" to beginning of sentence. (10-114)
	Paragraph 4.18.8.4.k.2	Add comma after word "cylindrical" in line 1 (10-114)
	Paragraph 4.18.8.5.c.5	Correct $S_{sb,1}$ with $S_{s,b,1}$ (10-114)
	Paragraph 4.18.8.8.2	Correct spelling for "integral" (10-114)
	Paragraph 4.18.8.8.3.b.3.i, ii	Correct $S_{PS,c}$ and $S_{PS,s}$ (10-114)
	Paragraph 4.18.8.8.3.d.1 , 2	Correct numbering and change item 2 to configuration a only(10-114)
	Paragraph 4.18.9.5	Move Paragraph 4.18.9.5 to 4.18.9.7
	Paragraph 4.18.9.7.2	Correct spelling for "integral" (10-114)
	Paragraph 4.18.9.7.3.b.3.i, ii	Correct $S_{PS,c}$ and $S_{PS,s}$ (10-114)
	Paragraph 4.18.9.7.3.d.1 , 2	Correct numbering (10-114)
	Nomenclature	Correct $S_{s,b,1}$ (10-114)
	Table 4.18.4	Various corrections (10-114)

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PART 1

GENERAL REQUIREMENTS

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1.1 General

1.1.1 Introduction

1.1.1.1 This Division contains mandatory requirements, specific prohibitions, and non-mandatory guidance for the design, materials, fabrication, examination, inspection, testing, and certification of pressure vessels and their associated pressure relief devices.

1.1.1.2 The Code does not address all aspects of these activities. Those aspects that are not specifically addressed should not be considered prohibited and shall be addressed by appropriate engineering judgment. Engineering judgment shall be consistent with the philosophy of this Division, and such judgments shall never be used to overrule mandatory requirements or specific prohibitions of this Division.

1.1.2 Organization

1.1.2.1 The requirements of this Division are contained in the nine Parts listed below. Each of these Parts and Annexes is composed of paragraphs that are identified by an alphanumeric numbering system in accordance with the ISO Standard Template for the Preparation of Normative-Type Documents. References to paragraphs are made directly by reference to the paragraph number. For example, the Scope is referenced as paragraph 1.2.

a) Part 1 – General Requirements, provides the scope of this division and establishes the extent of coverage

b) Part 2 – Responsibilities and Duties, sets forth the responsibilities of the user and Manufacturer, and the duties of the Inspector

c) Part 3 – Materials Requirements, provides the permissible materials of construction, applicable material specification and special requirements, physical properties, allowable stresses, and design fatigue curves

d) Part 4 – Design By Rule Requirements, provides requirements for design of vessels and components using rules

e) Part 5 – Design By Analysis Requirements, provides requirements for design of vessels and components using analytical methods

f) Part 6 – Fabrication Requirements, provides requirements governing the fabrication of vessels and parts

g) Part 7 – Examination and Inspection Requirements, provides requirements governing the examination and inspection of vessels and parts

h) Part 8 – Pressure Testing Requirements, provides pressure testing requirements

i) Part 9 – Pressure Vessel Overpressure Protection, provides rules for pressure relief devices

1.1.2.2 Mandatory and non-mandatory requirements are provided as normative and informative annexes, respectively, to the specific Part under consideration. The Normative Annexes address specific subjects not covered elsewhere in this Division and their requirements are mandatory when the subject covered is included in construction under this Division. Informative Annexes provide information and suggested good practices.

1.1.2.3 The materials, design, fabrication, examination, inspection, testing, and certification of pressure vessels and their associated pressure relief devices shall satisfy all applicable Parts and Normative Annexes shown above in order to qualify the construction in accordance with this Division.

1.1.3 Definitions

The definitions for the terminology used in this Part are contained in Annex 1.B.

1.2 Scope

1.2.1 Overview

1.2.1.1 In the scope of this division, pressure vessels are containers for the containment of pressure, either internal or external. This pressure may be obtained from an external source or by the application of heat from a direct or indirect source as a result of a process, or any combination thereof.

1.2.1.2 The rules of this Division may be used for the construction of the following pressure vessels.

a) Vessels to be installed at a fixed (stationary) location for a specific service where operation and maintenance control is retained during the useful life of the vessel by the user and is in conformance with the User's Design Specification required by Part 2.

b) Pressure vessels installed in ocean-going ships, barges, and other floating craft or used for motor vehicle or rail freight. For these applications it is necessary that prior written agreement with the jurisdictional authority be established covering operation and maintenance control for a specific service. This operation and maintenance control must be retained during the useful life of the pressure vessel by the user in conformance with the User's Design Specification required in Part 2. Such a pressure vessel as described above may be constructed and stamped within the scope of this Division provided it meets all other requirements as specified with the following additional provisions.

   1) Loading conditions imposed by movement of the pressure vessel during operation and by relocation of the pressure vessel between work sites or due to loading and discharge, as applicable, shall be considered in the design.

   2) The User's Design Specification shall include the agreements that define those aspects of operation and maintenance control unique to the particular pressure vessel.

c) Pressure vessels or parts subject to direct firing from the combustion of fuel (solid, liquid, or gaseous), that are not within the scope of Sections I, III, or IV may be constructed in accordance with the rules of this Division.

d) Unfired steam boilers shall be constructed in accordance with the rules of Section I or Section VIII, Division 1.

e) The following pressure vessels in which steam is generated shall be constructed in accordance with the rules of Section VIII, Division 1 or this Division:

   1) Vessels known as evaporators or heat exchangers;

   2) Vessels in which steam is generated by the use of heat resulting from operation of a processing system containing a number of pressure vessels such as used in the manufacture of chemical and petroleum products; and

   3) Vessels in which steam is generated but not withdrawn for external use.

1.2.1.3 The scope of this Division has been established to identify components and parameters considered in formulating the rules given in this Division. Laws or regulations issued by municipality, state, provincial, federal, or other enforcement or regulatory bodies having jurisdiction at the location of an installation establish the mandatory applicability of the Code rules, in whole or in part, within the jurisdiction. Those laws or regulations may require the use of this Division of the Code for vessels or components not considered to be within its scope. These laws or regulations should be reviewed to determine size or service limitations of the coverage which may be different or more restrictive than those given here.

1.2.2 Additional Requirements for Very High Pressure Vessels

1.2.2.1 The rules of this Division do not specify a limitation on pressure but are not all-inclusive for all types of construction. For very high pressures, some additions to these rules may be necessary to meet the design principles and construction practices essential to vessels for such pressures. However, only in the

event that, after application of additional design principles and construction practices, the vessel still complies with all of the requirements of the Code, may it be stamped with the Code symbol.

1.2.2.2 As an alternative to this Division, Section VIII, Division 3 should be considered for the construction of vessels intended for operating pressures exceeding 68.95 MPa (10,000 psi).

1.2.3 Geometric Scope of This Division

The scope of this Division is intended to include only the vessel and integral communicating chambers, and shall include the following:

a) Where external piping, other pressure vessels including heat exchangers, or mechanical devices (i.e. pumps, mixers, or compressors) are to be connected to the vessel:

   1) The welding end connection for the first circumferential joint for welded connections (see Table 4.2.4).

   2) The first threaded joint for screwed connections.

   3) The face of the first flange for bolted and flanged connections. Optionally, when the first flange is welded to the nozzle neck, the weld connecting the flange to the nozzle neck may be considered as the first circumferential joint, provided this construction is documented in the User's Design Specification and is properly described on the vessel drawing and the Manufacturer's Data Report Form.

   4) The first sealing surface for proprietary connections or fittings.

b) Where nonpressure parts are welded directly to either the internal or external pressure retaining surface of a pressure vessel, the scope of this Division shall include the design, fabrication, testing, and material requirements established for nonpressure part attachments by the applicable paragraphs of this Division (see paragraph 4.2.5.6).

c) Pressure retaining covers and their fasteners (bolts and nuts) for vessel openings, such as manhole and handhole covers.

d) The first sealing surface for proprietary connections, fittings or components that are designed to rules that are not provided by this Division, such as gages, instruments, and nonmetallic components.

1.2.4 Classifications Outside the Scope of this Division

1.2.4.1 The scope of this Division has been established to identify the components and parameters considered in formulating the rules given in this Division. Laws or regulations issued by a Jurisdictional Authority at the location of an installation establish the mandatory applicability of the Code rules, in whole or in part, within that jurisdiction. Those laws or regulations may require the use of this Division of the Code for vessels or components not considered to be within its Scope. These laws or regulations should be reviewed to determine size or service limitations that may be more restrictive than those given here.

1.2.4.2 The following vessels are not included in the scope of this Division. However, any pressure vessel, with the exception of (a) below, that is not excluded from the scope of this Division by paragraph 1.2.1.2 and that meets all applicable requirements of this Division may be stamped with the U2 Code symbol.

a) Vessels within the scope of other Sections.

b) Fired process tubular heaters as defined in API RP560.

c) Pressure containers that are integral parts or components of rotating or reciprocating mechanical devices, such as pumps, compressors, turbines, generators, engines, and hydraulic or pneumatic cylinders where the primary design considerations and/or stresses are derived from the functional requirements of the device.

d) Structures consisting of piping components, such as pipe, flanges, bolting, gaskets, valves, expansion joints, and fittings whose primary function is the transport of fluids from one location to another within a

system of which it is an integral part, that is, piping systems, including the piping system between a pressure relief device and the vessel it protects, see Part 9.

e) Pressure containing parts of components, such as strainers and devices that serve such purposes as mixing, separating, snubbing, distributing, and metering or controlling flow, provided that pressure containing parts of such components are generally recognized as piping components or accessories.

f) A vessel for containing water under pressure, including those containing air the compression of which serves only as a cushion, when none of the following limitations are exceeded:

   1) A design pressure of 2.07 MPa (300 psi)

   2) A design temperature of 99°C (210°F)

g) A hot water supply storage tank heated by steam or any other indirect means when none of the following limitations is exceeded:

   1) A heat input of 58.6 kW (200,000 Btu/hr)

   2) A water temperature of 99°C (210°F)

   3) A nominal water containing capacity of 454 liters (120 gal)

h) Vessels with an internal or external design pressure not exceeding 103 kPa (15 psi) with no limitation on size, for multi-chambered vessels, the design pressure on the common elements shall not exceed 103 kPa (15 psi).

i) Vessels with an inside diameter, width, height, or cross section diagonal not exceeding 150 mm (6 inches), with no limitation on length of vessel or pressure.

j) Pressure vessels for human occupancy (requirements for pressure vessels for human occupancy are covered in ASME PVHO-1).

1.2.5 Combination Units

When a pressure vessel unit consists of more than one independent pressure chamber, only the parts of chambers that come within the scope of this Division need be constructed in compliance with its provisions (see Part 4, paragraph 4.1.8).

1.2.6 Field Assembly of Vessels

1.2.6.1 Field assembly of vessels constructed to this Division may be performed as follows.

a) The Manufacturer of the vessel completes the vessel in the field, completes the Form A-1 Manufacturer's Data Report, and stamps the vessel.

b) The Manufacturer of parts of a vessel to be completed in the field by some other party stamps these parts in accordance with Code rules and supplies the Form A-2 Manufacturer's Partial Data Report to the other party. The other party, who must hold a valid U2 Certificate of Authorization, makes the final assembly, performs the required NDE, performs the final pressure test, completes the Form A-1 Manufacturer's Data Report, and stamps the vessel.

c) The field portion of the work is completed by a holder of a valid U2 Certificate of Authorization other than the vessel Manufacturer. The stamp holder performing the field work is required to supply a Form A-2 Manufacturer's Partial Data Report covering the portion of the work completed by his organization (including data on the pressure test if conducted by the stamp holder performing the field work) to the Manufacturer responsible for the Code vessel. The vessel Manufacturer applies his U2 Stamp in the presence of a representative from his Inspection Agency and completes the Form A-1 Manufacturer's Data Report with his Inspector.

1.2.6.2 In all three alternatives, the party completing and signing the Form A-1 Manufacturer's Data Report assumes full Code responsibility for the vessel. In all three cases, each Manufacturer's Quality Control System shall describe the controls to assure compliance by each Code stamp holder.

1.2.7 Pressure Relief Devices

The scope of this Division includes provisions for pressure relief devices necessary to satisfy the requirements of Part 9.

1.3 Standards Referenced by This Division

a) Throughout this Division, references are made to various standards, such as ASME/ANSI standards, which describe parts or fittings or which establish dimensional limits for pressure vessel parts. These standards, with the year of the acceptable edition, are listed in Table 1.1.

b) Rules for the use of these standards are stated elsewhere in this Division.

1.4 Units of Measurement

a) Either U.S. Customary, SI or any local customary units may be used to demonstrate compliance with all requirements of this edition (e.g. materials, design, fabrication, examination, inspection, testing, certification and overpressure protection).

b) A single system of units shall be used for all aspects of design except where unfeasible or impractical. When components are manufactured at different locations where local customary units are different than those used for the general design, the local units may be used for the design and documentation of that component. Similarly, for proprietary components or those uniquely associated with a system of units different than that used for the general design, the alternate units may be used for the design and documentation of that component.

c) For any single equation, all variables shall be expressed in a single system of units. When separate equations are provided for US Customary and SI units, those equations shall be executed using variables in the units associated with the specific equation. Data expressed in other units shall be converted to U.S. Customary or SI units for use in these equations. The result obtained from execution of these equations may be converted to other units.

d) Production, measurement and test equipment, drawings, welding procedure specifications, welding procedure and performance qualifications, and other fabrication documents may be in U.S. Customary, SI or local customary units in accordance with the fabricator's practice. When values shown in calculations and analysis, fabrication documents or measurement and test equipment are in different units, any conversions necessary for verification of Code compliance and to ensure that dimensional consistency is maintained shall be in accordance with the following:

   1) Conversion factors shall be accurate to at least four significant figures

   2) The results of conversions of units shall be expressed to a minimum of three significant figures

e) Conversion of units, using the precision specified above shall be performed to assure that dimensional consistency is maintained. Conversion factors between US Customary and SI units may be found in Annex 1.C. Whenever local customary units are used the Manufacturer shall provide the source of the conversion factors which shall be subject to verification and acceptance by the Authorized Inspector or Certified Individual.

f) Dimensions shown in the text, tables and figures, whether given as a decimal or a fraction, may be taken as a decimal or a fraction and do not imply any manufacturing precision or tolerance on the dimension.

g) Material that has been manufactured and certified to either the U.S. Customary or SI material specification (e.g. SA-516 or SA-516M) may be used regardless of the unit system used in design. Standard fittings (e.g. flanges, elbows, etc.) that have been certified to either US Customary units or SI units may be used regardless of the units system used in design.

h) All entries on a Manufacturer's Data Report and data for Code required nameplate marking shall be in units consistent with the fabrication drawings for the component using U. S. Customary, SI or local customary units. It is acceptable to show alternative units parenthetically. Users of this Code are cautioned that the receiving Jurisdiction should be contacted to ensure the units are acceptable.

1.5 Technical Inquires

A procedure for submittal of Technical Inquires to the ASME Boiler and Pressure Vessel Code Committee is contained in Annex 1.A.

1.6 Tables

Table 1.1 – Year Of Acceptable Edition Of Referenced Standards In This Division

Title	Number	Year
Unified Inch Screw Threads (UN and UNR Thread Form)	ASME B1.1	2003
Pipe Threads, General Purpose, Inch	ASME B1.20.1	1983 (R 2006) (1)
Pipe Flanges and Flanged Fittings	ASME B16.5	2003
Factory Made Wrought Steel Buttwelding Fittings	ASME B16.9	2007
Forged Steel Fittings, Socket-Welding and Threaded	ASME B16.11	2005
Metallic Gaskets for Pipe Flanges – Ring Joint, Spiral-Wound and Jacketed	ASME B16.20	2007
Large Diameter Steel Flanges (NPS 26 through NPS 60)	ASME B16.47	1996
Square and Hex Nuts (Inch Series)	ASME/ANSI B18.2.2	1987(R 1993) (1)
Welded and Seamless Wrought Steel Pipe	ANSI/ASME B36.10	2004
Pressure Relief Devices	ASME PTC 25	2001
Qualifications for Authorized Inspection	ASME QAI-1	2005
Seat Tightness of Pressure Relief Valves	API Standard 527	1991 (R 2002) (1)
ASNT Central Certification Program	ACCP	Rev. 3, 1997
ASNT Standard for Qualification and Certification of Nondestructive Testing Personnel	ANSI/ASNT CP-189	2006
Recommended Practice for Personnel Qualification and Certification in Nondestructive Testing	SNT-TC-1A	2006
Reference Photographs for Magnetic Particle Indications on Ferrous Casting	ASTM E 125	1963 (R 2003) (1)
Practice for Fabricating and Checking Aluminum Alloy Ultrasonic Standard Reference Blocks	ASTM E 127	1998
Hardness Conversion Tables for Metals	ASTM E 140	2002 - 2007
Standard Reference Radiographs for Heavy-Walled (2 to 4 1/2-in. (51 to 114-mm)) Steel Castings	ASTM E 186	1998
Method of Conducting Drop Weight Test to Determine Nil Ductility Transition Temperature of Ferritic Steel	ASTM E 208	2006
Reference Radiographs for High-Strength Copper-Base and Nickel-Copper Alloy Castings	ASTM E 272	1999
Standard Reference Radiographs for Heavy-Walled (4 1/2 to 12-in. (114 to 305-mm)) Steel Castings	ASTM E 280	1998
Standard Reference Radiographs for Steel Castings Up to 2 in. (51 mm) in Thickness	ASTM E 446	1998
Metric Screw Threads – M Profile	ASME B 1.13M	2001
Metric Screw Threads – MJ Profile	ASME B 1.21M	1997
Metric Heavy Hex Screws	ASME B 18.2.3.3M	1979 (R 2001) (1)
Metric Hex Bolts	ASME B 18.2.3.5M	1979 (R 2001) (1)
Metric Heavy Hex Bolts	ASME B 18.2.3.6M	1979 (R 2001) (1)
Metric Hex Nuts, Style 1	ASME B 18.2.4.1M	2002
Metric Hex Nuts, Style 2	ASME B 18.2.4.2M	1979 (R 1995) (1)
Metric Heavy Hex Nuts	ASME B 18.2.46M	1979 (R 2003) (1)
Fitness-For-Service	API 579-1/ASME FFS-1	2007
Guidelines for Pressure Boundary Bolted Flange Joint Assembly	ASME PCC-1	2000

Table 1.1 – Year Of Acceptable Edition Of Referenced Standards In This Division

Title	Number	Year
Standard Procedures for Calibrating Magnetic Instruments to Measure the Delta Ferrite Content of Austenitic and Duplex Austenitic-Ferrite Stainless Steel Weld Metal	AWS 4.2	1997
Metallic materials - Charpy pendulum impact test - Part 1: Test method	ISO 148-1	2006
Metallic materials - Charpy pendulum impact test - Part 2: Verification of test machines	ISO 148-2	1998
Metallic materials - Charpy pendulum impact test - Part 3: Preparation and characterization of Charpy V reference test pieces for verification of test machines	ISO 148-3	1998
Note: 1) R indicates reaffirmed		

ANNEX 1.A

SUBMITTAL OF TECHNICAL INQUIRIES TO THE BOILER AND PRESSURE VESSEL STANDARDS COMMITTEE

(NORMATIVE)

1.A.1 Introduction

1.A.1.1 This Appendix provides guidance to Code users for submitting technical inquiries to the Committee. Technical inquiries include requests for revisions or additions to the Code rules, requests for Code Cases, and requests for Code interpretations as described below. Additional requirements for technical inquiries involving the addition of new materials to the Code are covered in Guideline on the Approval of New Materials under the ASME Boiler and Pressure Vessel Code, Section II, Parts C and D.

a) Code Revisions – Code revisions are considered to accommodate technological developments, address administrative requirements, incorporate Code Cases, or to clarify Code intent .

b) Code Cases – Code Cases represent alternatives or additions to existing Code rules. Code Cases are written as a question and reply and are usually intended to be incorporated into the Code at a later date. When used, Code Cases prescribe mandatory requirements in the same sense as the text of the Code. However, users are cautioned that not all Jurisdictions or owners automatically accept Code Cases. The most common applications for Code Cases are:

   1) to permit early implementation of an approved Code revision based on an urgent need;

   2) to permit the use of a new material for Code construction;

   3) to gain experience with new materials or alternative rules prior to incorporation directly into the Code.

c) Code Interpretations – Code Interpretations provide clarification of the meaning of existing rules in the Code and are also presented in question and reply format. Interpretations do not introduce new requirements. In cases where existing Code text does not fully convey the meaning that was intended, and revision of the rules is required to support an interpretation, an Intent Interpretation will be issued, and the Code will be revised.

1.A.1.2 The Code rules, Code Cases, and Code interpretations established by the Committee are not to be considered as approving, recommending, certifying, or endorsing any proprietary or specific design or as limiting in any way the freedom of manufacturers, constructors, or owners to choose any method of design or any form of construction that conforms to the Code rules.

1.A.1.3 Inquiries that do not comply with the provisions of this Appendix or that do not provide sufficient information for the Committee's full understanding may result in the request being returned to the inquirer with no action.

1.A.2 Inquiry Format

Submittals to the Committee shall include:

a) *Purpose.* Specify one of the following:

   1) revision of present Code rules;

   2) new or additional Code rules;

   3) Code Case;

   4) Code interpretation.

b) *Background.* Provide the information needed for the Committee's understanding of the inquiry, being sure to include reference to the applicable Code Section, Division, Edition, Addenda, paragraphs, figures, and tables. Preferably, provide a copy of the specific referenced portions of the Code.

c) *Presentations.* The inquirer may desire or be asked to attend a meeting of the Committee to make a formal presentation or to answer questions from the Committee members with regard to the inquiry. Attendance at a Committee meeting shall be at the expense of the inquirer. The inquirer's attendance or lack of attendance at a meeting shall not be a basis for acceptance or rejection of the inquiry by the Committee.

1.A.3 Code Revisions or Additions

Requests for Code revisions or additions shall provide the following.

a) *Proposed Revision or Additions.* For revisions, identify the rules of the Code that require revision and submit a copy of the appropriate rules as they appear in the Code marked up with the proposed revision. For additions, provide the recommended wording referenced to the existing Code rules.

b) *Statement of Need.* Provide a brief explanation of the need for the revision or addition.

c) *Background Information.* Provide background information to support the revision or addition including any data or changes in technology that form the basis for the request that will allow the Committee to adequately evaluate the proposed revision or addition. Sketches, tables, figures, and graphs should be submitted as appropriate. When applicable, identify any pertinent paragraph in the Code that would be affected by the revision or addition and identify paragraphs in the Code that reference the paragraphs that are to be revised or added.

1.A.4 Code Cases

Requests for Code Cases shall provide a *Statement of Need* and *Background Information* similar to that defined in paragraph 1.A.3.b and 1.A.3.c, respectively, for Code revisions or additions. The urgency of the Code Case (e.g., project underway or imminent, new procedure, etc.) shall be defined and it shall be confirmed that the request is in connection with equipment that will be ASME stamped, with the exception of Section XI applications. The proposed Code Case should identify the Code Section and Division and be written as a *Question* and a *Reply* in the same format as existing Code Cases. Requests for Code Cases should also indicate the applicable Code Editions and Addenda to which the proposed Code Case applies.

1.A.5 Code Interpretations

a) Requests for Code interpretations shall provide the following.

   1) *Inquiry.* Provide a condensed and precise question, omitting superfluous background information, and, when possible, composed in such a way that a “yes” or a “no” *Reply* with brief provisos if needed, is acceptable. The question should be technically and editorially correct.

   2) *Reply.* Provide a proposed *Reply* that will clearly and concisely answer the *Inquiry* question. Preferably, the *Reply* should be “yes” or “no” with brief provisos, if needed.

   3) *Background Information.* Provide any background information that will assist the Committee in understanding the proposed *Inquiry* and *Reply.*

b) Requests for Code interpretations shall be limited to an interpretation of a particular requirement in the Code or a Code Case. The Committee cannot consider consulting type requests such as the following:

   1) A review of calculations, design drawings, welding qualifications or descriptions of equipment or parts to determine compliance with Code requirements.

   2) A request for assistance in performing any Code prescribed functions relating to, but not limited to material selection, designs, calculations, fabrication, inspection, pressure testing or installation.

   3) A request seeking the rationale for Code requirements.

1.A.6 Submittals

Submittals to and responses from the Committee shall meet the following:

a) Submittal. Inquiries from Code users shall be in English and preferably be submitted in typewritten form; however, legible handwritten inquiries will also be considered. They shall include the name, address, telephone number, fax number and e-Mail address, if available, of the inquirer and be mailed to the following address:

   Secretary
   ASME Boiler and Pressure Vessel Committee
   Three Park Avenue
   New York, NY 10016-5990

   As an alternative, inquiries may be submitted via e-mail to: SecretaryBPV@asme.org

b) *Response.* The Secretary of the ASME Boiler and Pressure Vessel Committee or of the appropriate Subcommittee shall acknowledge receipt of each properly prepared inquiry and shall provide a written response to the inquirer upon completion of the requested action by the Code Committee.

ANNEX 1.B

DEFINITIONS

(NORMATIVE)

1.B.1 Introduction

This Annex contains definitions of terms generally used in this Division. Definitions relating to specific applications may also be found in related Parts of this Division.

1.B.2 Definition of Terms

1.B.2.1 Acceptance by the Inspector, accepted by the Inspector - an indication that the Inspector has reviewed a subject in accordance with his duties as required by the rules of this Division and after such review is able to sign the Certificate of Inspection for the applicable Manufacturer's Data Report Form.

1.B.2.2 ASME Designated Organization – an entity authorized by ASME to perform administrative functions on its behalf.

1.B.2.3 ASME Designee – an individual authorized by ASME to perform administrative functions on its behalf.

1.B.2.4 Certificate of Compliance – a document that states that the material represented has been manufactured, sampled, tested and inspected in accordance with the requirements of the material specification (including year of issue) and any other requirements specified in the purchase order or contract shown on the certificate and has been found to meet such requirements. This document may be combined with the Materials Test Report (see 1.B.2.11) as a single document.

1.B.2.5 Communicating Chambers – appurtenances to a vessel that intersect the shell or heads of a vessel and form an integral part of the pressure containing enclosure.

1.B.2.6 Construction – an all-inclusive term comprising materials, design, fabrication, examination, inspection, testing, certification, and pressure relief.

1.B.2.7 Local Jurisdictional Authority – an agency enforcing laws or regulations applicable to pressure vessels.

1.B.2.8 Manufacturer – the organization responsible for construction of a pressure vessel, vessel component, or part or the organization responsible for the manufacture of pressure relief devices in accordance with the rules of this Division and who holds an ASME Certificate of Authorization to apply the Code Symbol Stamp to such an item.

1.B.2.9 Material – any substance or product form covered by a material specification in Section II Parts A, B, or C or any other substance or product form permitted for use in pressure vessel construction by this Division.

1.B.2.10 Material Manufacturer – the organization responsible for the production of products meeting the requirements of the material specification and accepting the responsibility for any statements or data in any required Certificate of Compliance or Material Test Report representing the material.

1.B.2.11 Material Test Report – a document in which are recorded the results of tests, examinations, repairs, or treatments required by the material specification to be reported. Including those of any supplementary requirements or other requirements stated in the order for the material. This document may be combined with a Certificate of Compliance (see 1.B.2.4) as a single document. When preparing a material test report, a material manufacturer may transcribe data produced by other organizations provided he accepts responsibility for the accuracy and authenticity of the data.

1.B.2.12 User – the organization that purchases the finished pressure vessel for its own use or as an agent for the owner. The user's designated agent may be either a design agency specifically engaged by the user, the Manufacturer of a system for a specific service which includes a pressure vessel as a part and which is purchased by the user, or an organization which offers pressure vessels for sale or lease for specific services.

ANNEX 1.C

GUIDANCE FOR THE USE OF US CUSTOMARY AND SI UNITS IN THE ASME BOILER AND PRESSURE VESSEL CODES

(INFORMATIVE)

1.C.1 Use of Units in Equations

The equations in this Division are suitable for use only with either the SI or US customary units provided in this Annex, or with the units provided in the nomenclature associated with that equation. It is the responsibility of the individual and organization performing the calculations to ensure that appropriate units are used. Either SI or US Customary units may be used as a consistent set. When necessary to convert from one system to another, the units shall be converted to at least four significant figures for use in calculations and other aspects of construction.

1.C.2 Guidelines Used to Develop SI Equivalents

a) US Customary units are placed in parenthesis after the SI unit in the text.

b) In general, both SI and US Customary tables are provided if interpolation is expected. The table designation (e.g. table number) is the same for both the SI and the US Customary tables, with the addition of an M after the table number for the SI Table. In the text, references to a Table use only the primary table number (i.e. without the M). For some small tables, where interpolation is not required, US Customary units are placed in parenthesis after the SI unit.

c) Separate SI and US Customary versions of graphical information (charts) are provided, except that if both axes are dimensionless a single figure (chart) is used.

d) In most cases, conversions of units in the text were done using hard SI conversion practices, with some soft conversions on a case-by-case basis as appropriate. This was implemented by rounding the SI values to the number of significant figures of implied precision in the existing US customary units. For example, 3000 psi has an implied precision of one significant figure. Therefore, the conversion to SI units would typically be to 20,000 kPa. This is a difference of about 3% from the "exact" or soft conversion of 20684.27 kPa. However, the precision of the conversion was determined by the Committee on a case-by-case basis. More significant digits were included in the SI equivalent if there was any question. The values of allowable stress in Section II, Part D generally include 3 significant figures.

e) Minimum thickness and radius values that are expressed in fractions of an inch were generally converted according to Table 1.C.1.

f) For nominal sizes that are in even increments of inches, even multiples of 25 mm were generally used. Intermediate values were interpolated rather than converting and rounding to the nearest mm. See examples in Table 1.C.2. Note that this table does not apply to nominal pipe sizes (NPS), which are covered in Table 1.C.4

g) For nominal pipe sizes, the relationships shown in Table 1.C.4 were used.

h) Areas in square inches (in^2) were converted to square mm (mm^2) and areas in square feet (ft^2) were converted to square meters (m^2), see examples in Table 1.C.5.

i) Volumes in cubic inches (in^3) were converted to cubic mm (mm^3) and volumes in cubic feet (ft^3) were converted to cubic meters (m^3), see examples in the Table 1.C.6.

j) Although the pressure should always be in MPa or psi for calculations, there are cases where other units are used in the text. For example, kPa is sometimes used for low pressures and ksi is sometimes used for high pressures and stresses. Also, rounding was to one significant figure (two at the most) in most cases, see examples in Table 1.C.7. Note that 14.7 psi converts to 101 kPa, while 15 psi converts to 100 kPa. While this may seem at first glance to be an anomaly, it is consistent with the rounding philosophy.

k) Material properties that are expressed in psi or ksi (e.g. allowable stress, yield and tensile strength, elastic modulus) were generally converted to MPa to three significant figures. See example in Table 1.C.8.

l) In most cases, temperatures (e.g. for PWHT) were rounded to the nearest 5°C. Depending on the implied precision of the temperature, some were rounded to the nearest 1°C or 10°C or even 25°C. Temperatures colder than 0°F (negative values) were generally rounded to the nearest 1°C. The examples in Table 1.C.9 were created by rounding to the nearest 5°C, with one exception.

1.C.3 Soft Conversion Factors

Table 1.C.10 of "soft" conversion factors is provided for convenience. Multiply the US Customary value by the factor given to obtain the SI value. Similarly, divide the SI value by the factor given to obtain the US Customary value. In most cases it is appropriate to round the answer to three significant figures.

1.C.4 Tables

Table 1.C.1 – Typical Size or Thickness Conversions for Fractions

Fraction in US Customary Units	Proposed SI Conversion	Difference
1/32 inch	0.8 mm	-0.8%
3/64 inch	1.2 mm	-0.8%
1/16 inch	1.5 mm	5.5%
3/32 inch	2.5 mm	-5.0%
1/8 inch	3 mm	5.5%
5/32 inch	4 mm	-0.8%
3/16 inch	5 mm	-5.0%
7/32 inch	5.5 mm	1.0%
1/4 inch	6 mm	5.5%
5/16 inch	8 mm	-0.8%
3/8 inch	10 mm	-5.0%
7/16 inch	11 mm	1.0%
1/2 inch	13 mm	-2.4%
9/16 inch	14 mm	2.0%
5/8 inch	16 mm	-0.8%
11/16 inch	17 mm	2.6%
3/4 inch	19 mm	0.3%
7/8 inch	22 mm	1.0%
1 inch	25 mm	1.6%

Table 1.C.2 – Typical Size or Thickness Conversions

Size – inches	Size – mm
1	25
1-1/8	29
1-1/4	32
1-1/2	38
2	50
2-1/4	57
2-1/2	64
3	75
3-1/2	89
4	100
4-1/2	114
5	125
6	150
8	200
12	300
18	450
20	500
24	600
36	900
40	1000
54	1350
60	1500
72	1800

Table 1.C.3– Typical Size or Length Conversions

Size or Length	Size or Length
3 ft	1 m
5 ft	1.5 m
200 ft	60 m

Table 1.C.4 – Typical Nominal Pipe Size Conversions

U.S. Customary Practice	SI Practice	U. S. Customary Practice	SI Practice
NPS 1/8	DN 6	NPS 20	DN 500
NPS ¼	DN 8	NPS 22	DN 550
NPS 3/8	DN 10	NPS 24	DN 600
NPS ½	DN 15	NPS 26	DN 650
NPS ¾	DN 20	NPS 28	DN 700
NPS 1	DN 25	NPS 30	DN 750
NPS 1-1/4	DN 32	NPS 32	DN 800
NPS 1-1/2	DN 40	NPS 34	DN 850
NPS 2	DN 50	NPS 36	DN 900
NPS 2-1/2	DN 65	NPS 38	DN 950
NPS 3	DN 80	NPS 40	DN 1000
NPS 3-1/2	DN 90	NPS 42	DN 1050
NPS 4	DN 100	NPS 44	DN 1100
NPS 5	DN 125	NPS 46	DN 1150
NPS 6	DN 150	NPS 48	DN 1200
NPS 8	DN 200	NPS 50	DN 1250
NPS 10	DN 250	NPS 52	DN 1300
NPS 12	DN 300	NPS 54	DN 1350
NPS 14	DN 350	NPS 56	DN 1400
NPS 16	DN 400	NPS 58	DN 1450
NPS 18	DN 450	NPS 60	DN 1500

Table 1.C.5 – Typical Area Conversions

Area – US Customary	Area - SI
1 in^2	650 mm^2
6 in^2	4,000 mm^2
10 in^2	6,500 mm^2
5 ft^2	0.5 m^2

Table 1.C.6 – Typical Volume Conversions

Volume – US Customary	Volume - SI
1 in^3	16,000 mm^3
6 in^3	100,000 mm^3
10 in^3	160,000 mm^3
5 ft^3	0.14 m^3

Table 1.C.7 – Typical Pressure Conversions

Pressure– US Customary	Pressure - SI
0.5 psi	3 kPa
2 psi	15 kPa
3 psi	20 kPa
10 psi	70 kPa
14.7 psi	101 kPa
15 psi	100 kPa
30 psi	200 kPa
50 psi	350 kPa
100 psi	700 kPa
150 psi	1 MPa
200 psi	1.5 MPa
250 psi	1.7 MPa
300 psi	2 MPa
350 psi	2.5 MPa
400 psi	3 MPa
500 psi	3.5 MPa
600 psi	4 MPa
1,200 psi	8 MPa
1,500 psi	10 MPa

Table 1.C.8 – Typical Strength Conversions

Strength – US Customary	Strength - SI
30,000 psi	205 MPa
38,000 psi	260 MPa
60,000 psi	415 MPa
70,000 psi	480 MPa
95,000 psi	655 MPa

Table 1.C.9 – Typical Temperature Conversions

Temperature - °F	Temperature °C
70	20
100	38
120	50
150	65
200	95
250	120
300	150
350	175
400	205
450	230
500	260
550	290
600	315
650	345
700	370
750	400
800	425
850	455
900	480
925	495
950	510
1000	540
1050	565
1100	595
1150	620
1200	650
1250	675
1800	980
1900	1040
2000	1095
2050	1120

Table 1.C.10 – Conversion Factors

US Customary	SI	Conversion Factor	Notes
in	mm	25.4	---
ft	m	0.3048	---
in^2	mm^2	645.16	---
ft^2	m^2	0.09290304	---
in^3	mm^3	16,387.064	---
ft^3	m^3	0.02831685	---
US Gal.	m^3	0.003785412	---
psi	MPa	0.0068948	Used exclusively in equations
psi	kPa	6.894757	Used only in text and for nameplate
ft-lbs	J	1.355818	---
°F	°C	5/9(°F – 32)	Not for temperature difference
°F	°C	5/9(°F)	For temperature differences only
R	K	5/9	Absolute temperature
lbm	kg	0.4535924	---
lbf	N	4.448222	---
in-lbs	N-mm	112.98484	Use exclusively in equations
ft-lbs	N-m	1.3558181	Use only in text
$ksi\sqrt{in}$	$MPa\sqrt{m}$	1.0988434	---
BTU/hr	W	0.2930711	Use for Boiler rating and heat transfer
lbs/ft^3	kg/m^3	16.018463	---

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PART 2

RESPONSIBILITIES AND DUTIES

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2.1 General

2.1.1 Introduction

The user, Manufacturer, and Inspector involved in the production and certification of vessels in accordance with this Division have definite responsibilities or duties in meeting the requirements of this Division. The responsibilities and duties set forth in the following relate only to compliance with this Division, and are not to be construed as involving contractual relations or legal liabilities.

2.1.2 Definitions

The definitions for the terminology used in this Part are contained in Annex 1.B.

2.1.3 Code Reference

The Code Edition year and Addenda Date on the User's Design Specification and Manufacturer's Design Report shall be the same as the Code Edition year and Addenda Date on the Manufacturer's Data Report.

2.2 User Responsibilities

2.2.1 General

It is the responsibility of the user or an agent acting on behalf of the user to provide a certified User's Design Specification for each pressure vessel to be constructed in accordance with this Division. The User's Design Specification shall contain sufficient detail to provide a complete basis for design and construction in accordance with this Division. It is the user's responsibility to specify, or cause to be specified, the effective Code edition and Addenda to be used for construction.

2.2.2 User's Design Specification

2.2.2.1 The User's Design Specification shall include but not necessarily be limited to the following:

a) Installation Site
   1) Location
   2) Jurisdictional authority if applicable
   3) Environmental conditions
      i) Wind design loads including relevant factors (i.e. design wind speed, exposure, gust factors)
      ii) Earthquake design loads
      iii) Snow loads
      iv) Lowest one day mean temperature for location

b) Vessel Identification
   1) Vessel number or identification
   2) Service fluid for proprietary fluids specific properties needed for design, e.g., gas, liquid, density, etc.

c) Vessel Configuration and Controlling Dimensions

1) Outline drawings

2) Vertical or horizontal

3) Openings, connections, closures including quantity, type and size, and location (i.e. elevation and orientation)

4) Principal component dimensions in sufficient detail so that volume capacities can be determined

5) Support method

d) Design Conditions

1) Specified design pressure. The specified design pressure is the design pressure, see paragraph 4.1.5.2.a, required at the top of the vessel in its operating position. It shall include suitable margins required above the maximum anticipated operating pressure to ensure proper operation of the pressure relief devices. The MAWP of the vessel may be set equal to this specified design pressure. If the actual MAWP of the vessel is calculated, it shall not be less than the specified design pressure.

2) Design temperature and coincident specified design pressure (see paragraph 4.1.5.2.d).

3) Minimum Design Metal Temperature (MDMT) and coincident specified design pressure (see paragraph 4.1.5.2.e).

4) Dead loads, live loads and other loads required to perform the load case combinations required in Parts 4 and 5.

e) Operating Conditions

1) Operating pressure

2) Operating temperature

3) Fluid transients and flow and sufficient properties for determination of steady state and transient thermal gradients across the vessel sections, if applicable (see paragraph 5.5.2)

f) Design Fatigue Life

1) Cyclic operating conditions and whether or not a fatigue analysis of the vessel as required shall be determined in accordance with paragraph 4.1.1.4. When a fatigue analysis is required, provide information in sufficient detail so that an analysis of the cyclic operation can be carried out in accordance with paragraph 5.5.

2) When a vessel is designed for cyclic conditions, the number of design cycles per year and the required vessel design life in years shall be stated.

3) When cyclic operating conditions exist and a fatigue analysis is not required based on comparable equipment experience, this shall be stated. The possible harmful effects of the design features listed in 5.5.2.2.a) through f) shall be evaluated when contemplating comparable equipment experience.

g) Materials of Construction

1) Material specification requirements shall be in accordance with one or more of the following criteria.

i) Specification of materials of construction in accordance with Part 3.

ii) Generic material type (i.e. carbon steel or Type 304 Stainless Steel). The user shall specify requirements that provide an adequate basis for selecting materials to be used for the construction of the vessel. The Manufacturer shall select the appropriate material from Part 3, considering information provided by the user per paragraph 2.2.2.1.g.3.

2) The user shall specify the corrosion and/or erosion allowance.

3) The user, when selecting the materials of construction, shall consider the following:

i) Damage mechanisms associated with the service fluid at design conditions. Informative and non-mandatory guidance regarding metallurgical phenomena is provided in Section II, Part D, Appendix A, API RP 571, and WRC Bulletins 488, 489 and 490.

ii) Minimum Design Metal Temperature and any additional toughness requirements.

iii) The need for specific weld filler material to meet corrosion resistance requirements, see paragraph 6.2.5.8.

h) Loads and Load Cases

1) The user shall specify all expected loads and load case combinations as listed in paragraph 4.1.5.3.

2) These loading data may be established by:

i) Calculation

ii) Experimental methods

iii) Actual experience measurement from similar units

iv) Computer analysis

v) Published data

i) Overpressure Protection

1) The user shall be responsible for the design, construction and installation of the overpressure protection system unless it is delegated to the Manufacturer. This system shall meet the requirements of Part 9.

2) The type of over pressure protection intended for the vessel shall be documented in the User's Design Specification as follows (see paragraph 9.1):

i) Type of overpressure protection system (e.g., type of pressure relief valve, rupture disc, etc.)

ii) System design (see paragraph 9.7)

3) The user shall state if jurisdictional acceptance is required prior to operation of the vessel.

2.2.2.2 Additional Requirements – The user shall state what additional requirements are appropriate for the intended vessel service such as:

a) Additional requirements such as non-destructive examination, restricted chemistry, or heat treatments

b) Type of weld joints and the extent of required nondestructive examinations

c) Non-mandatory or optional provisions of this Division that are considered to be mandatory for the subject vessel

d) Any special requirements for marking and their location (see paragraph 4.1 and Annex 2.F)

e) Requirements for seals and/or bolting for closures and covers

f) Additional requirements relating to erection loadings

g) Any agreements which resolve the problems of operation and maintenance control unique to the particular pressure vessel.

h) Specific additional requirements relating to pressure testing such as:

1) Fluid properties and test temperature limits

2) Position of vessel and support/foundation adequacy if field hydrostatic testing is required

3) Location: Manufacturer's facility or on-site

4) Cleaning and drying

5) Selection of pressure test method, see paragraph 8.1.1

6) Application of paints, coatings and linings, see paragraph 8.1.2.e

2.2.2.3 The User's Design Specification shall be certified in accordance with Annex 2.A.

2.3 Manufacturer's Responsibilities

2.3.1 Code Compliance

2.3.1.1 The Manufacturer is responsible for the structural and pressure retaining integrity of a vessel or part thereof, as established by conformance with the requirements of the rules of this Division and the requirements in the User's Design Specification.

2.3.1.2 The Manufacturer completing any vessel or part marked with the U2 symbol in accordance with this Division has the responsibility to comply with all the applicable requirements of this Division and, through proper certification, to ensure that any work by others also complies with the requirements of this Division. The Manufacturer shall certify compliance with these requirements by completing a Manufacturer's Data Report (see paragraph 2.3.4).

2.3.2 Materials Selection

2.3.2.1 When generic material types (i.e. carbon steel or Type 304 Stainless Steel) are specified, the Manufacturer shall select the appropriate material from Part 3, considering information provided by the user per paragraph 2.2.2.1 g.3).

2.3.2.2 Any material substitutions by the Manufacturer are subject to approval of the user.

2.3.3 Manufacturer's Design Report

2.3.3.1 The Manufacturer shall provide a Manufacturer's Design Report that includes:

a) Final as-built drawings.

b) The actual material specifications used for each component.

c) Design calculations and analysis that establish that the design as shown on the drawings complies with the requirements of this Division for the design conditions that have been specified in the User's Design Specification.

   1) Documentation of design-by-rule calculations in Part 4 shall include the following:

      i) The name and version of computer software, if applicable

      ii) Loading conditions and boundary conditions used to address the load cases in the User's Design Specification

      iii) Material models utilized for all required physical properties (i.e. stress-strain data, modulus of elasticity, Poisson's ratio, thermal expansion coefficient, thermal conductivity, thermal diffusivity), strength parameters (i.e. yield and tensile strength), and allowable stresses

      iv) Detailed calculations, including results from all of the applicable steps in the calculations, showing the acceptance criteria utilized to meet the requirements of this Division.

      v) A summary of the calculation results

   2) Documentation of design-by-analysis calculations in Part 5 shall include the following:

i) A detailed description of the numerical method used, including the name and version of computer software, if applicable

ii) Description of model geometry (including element type for finite element analysis)

iii) Loading conditions and boundary conditions used to address the load cases in the User's Design Specification

iv) Material models utilized for all required physical properties (i.e. modulus of elasticity, Poisson's ratio, thermal expansion coefficient, thermal conductivity, thermal diffusivity), strength parameters (i.e. yield and tensile strength), strain limits, if applicable, and the design membrane stress intensity per Part 3

v) Description of whether material nonlinearity is utilized in the analysis including a description of the material model (i.e. stress-strain curve and cyclic stress-strain curve)

vi) Description of the numerical analysis procedure (i.e. static analysis, thermal analysis (temperature and stress), buckling analysis, natural frequency analysis, dynamic analysis) and whether a geometrically linear or nonlinear option is invoked

vii) Graphical display of relevant results (i.e. numerical model, deformed plots, and contour plots of thermal and stress results)

viii) Method used to validate the numerical model (i.e. mesh sensitivity review and equilibrium check for finite element analysis, e.g. check of hoop stress in a component away from structural discontinuity and a check to ensure that global equilibrium is achieved between applied loads and reactions at specified boundary conditions)

ix) Description of results processing performed to establish numerical analysis results (i.e. stress linearization method, use of centroidal or nodal values for stress, strain, and temperature results)

x) A summary of the numerical analysis results showing the acceptance criteria utilized to meet the requirements of this Division

xi) Electronic storage of analysis results including input files and output files that contain numerical analysis results utilized to demonstrate compliance with the requirements of this Division

d) The results of any fatigue analyses according to paragraph 5.5, as applicable.

e) Any assumptions used by the Manufacturer to perform the vessel design.

2.3.3.2 The Manufacturer's Design Report shall be certified in accordance with Annex 2.B.

2.3.4 Manufacturer's Data Report

The Manufacturer shall certify compliance to the requirements of this Division by the completion of the appropriate Manufacturer's Data Report as described in Annex 2.C and Annex 2.D.

2.3.5 Manufacturer's Construction Records

The Manufacturer shall prepare, collect and maintain construction records and documentation as fabrication progresses, to show compliance with the Manufacturer's Design Report (e.g., NDE reports, repairs, deviations from drawings, etc.) An index of the construction records files, in accordance with the Manufacturer's Quality Control system, shall be maintained current (see paragraph 2.C.3). These construction records shall be maintained by the Manufacturer for the duration as specified in paragraph 2.C.3.

2.3.6 Quality Control System

The Manufacturer shall have and maintain a Quality Control System in accordance with Annex 2.E.

2.3.7 Certification of Subcontracted Services

2.3.7.1 The Quality Control system shall describe the manner in which the Manufacturer (Certificate Holder) controls and accepts the responsibility for the subcontracting of activities. The Manufacturer shall ensure that all contracted activities meet the requirements of this Division.

2.3.7.2 Work such as forming, nondestructive examination, heat treating, etc., may be performed by others (for welding, see paragraph 6.1.4.2). It is the vessel Manufacturer's responsibility to ensure that all work performed complies with all the applicable requirements of this Division. After ensuring compliance, and obtaining concurrence of the Inspector, the vessel may be stamped with the ASME symbol.

2.3.7.3 Subcontracts that involve welding on the pressure boundary components for construction under the rules of this Division, other than as provided in paragraph 6.1.4.2 and for repair welds permitted by the ASME material specifications, shall be made only to subcontractors holding a valid U2 Certificate of Authorization. All such subcontracted welding shall be documented on the Form A-2, see Annex 2.D.

2.3.7.4 A Manufacturer may engage individuals by contract for their services as Welders or Welding Operators, at shop or site locations shown on his Certification of Authorization, provided all of the following conditions are met:

a) The work to be done by Welders or Welding Operators is within the scope of the Certificate of Authorization.

b) The use of such Welders or Welding Operators is described in the Quality Control system of the Manufacturer. The Quality Control System shall include a requirement for direct supervision and direct technical control of the Welders and Welding operators, acceptable to the Manufacturer's accredited Authorized Inspection Agency.

c) The Welding Procedures have been properly qualified by the Manufacturer, according to Section IX.

d) The Welders and Welding Operators are qualified by the Manufacturer according to Section IX to perform these procedures.

e) Code responsibility and control is retained by the Manufacturer.

2.3.8 Inspection and Examination

The Manufacturer's responsibility for inspection and examination is summarized in Annex 7.A.

2.3.9 Application of Code Stamp

Vessels or parts shall be stamped in accordance with the requirements in Annex 2.F. The procedure to obtain and use a Code Stamp is described in Annex 2.G.

2.4 The Inspector

2.4.1 Identification of Inspector

All references to Inspectors throughout this Division mean the Authorized Inspector as defined in this paragraph. All inspections required by this Division shall be by an Inspector regularly employed by an ASME accredited Authorized Inspection Agency or by a company that manufacturers pressure vessels exclusively for its own use and not for resale that is defined as a User-Manufacturer. This is the only instance in which an Inspector may be in the employ of the Manufacturer.

2.4.2 Inspector Qualification

All Inspectors shall have been qualified by a written examination under the rules of any state of the United States, province of Canada, or other jurisdiction, that has adopted the Code.

2.4.3 Inspector's Duties

2.4.3.1 It is the duty of the Inspector to make all the inspections specified by the rules of this Division. In addition, the Inspector shall make other such inspections as considered necessary in order to ensure that all requirements have been met. Some typical required inspections and verifications that are defined in the applicable rules are included in the Inspector's responsibility for inspection and examination as summarized in Annex 7A and verification that the Manufacturer has a valid Certificate of Authorization and is working according to an approved Quality Control System.

2.4.3.2 The Inspector of the completed vessel does not have the duty of establishing the accuracy of the design analysis but has the duty of establishing that the required analysis has been performed. The Inspector has the duty of verifying that the Manufacturer of the completed vessel has the User's Design Specification on file and that the requirements specified therein have been addressed in the Manufacturer's Design Report. The Inspector shall verify that both the User's Design Specification and the Manufacturer's Design Report are certified in accordance with the requirements of this Division.

2.4.3.3 The Inspector shall verify that the Manufacturer has a valid Certificate of Authorization and is working according to an approved Quality Control System including having a system in place to maintain the documentation for the Manufacturer's construction records current with production, and the reconciliation of any deviations from the Manufacturer's Design Report.

2.4.3.4 The Inspector shall certify the Manufacturer's Data Report. When the Inspector has certified by signing the Manufacturer's Data Report, this indicates acceptance by the Inspector. This acceptance does not imply assumption by the Inspector of any responsibilities of the Manufacturer.

ANNEX 2.A

GUIDE FOR CERTIFYING A USER'S DESIGN SPECIFICATION

(NORMATIVE)

2.A.1 General

An individual(s) in responsible charge of the specification of the vessel and the required design conditions shall certify that the User's Design Specification meets the requirements of this Division and any additional requirements needed for adequate design. Such certification requires the signature(s) of one or more Engineers with requisite experience and qualifications as defined below. One or more individuals may sign the documentation based on information they reviewed, and the knowledge and belief that the objectives of this Division have been satisfied.

2.A.2 Certification of the User's Design specification

2.A.2.1 One or a combination of methods shown below shall be used to certify the User's Design specification.

a) One or more Professional Engineers, registered in one or more of the states of the United States of America or the provinces of Canada and experienced in pressure vessel design, shall certify that the User's Design Specification meets the requirements in paragraph 2.2.2, and shall apply the Professional Engineer seal in accordance with the required procedures. In addition, the Registered Professional Engineer(s) shall prepare a statement to be affixed to the document attesting to compliance with the applicable requirements of the Code (see paragraph 2.A.2.5). This Professional Engineer shall be other than the Professional Engineer who certifies the Manufacturer's Design Report, although both may be employed by or affiliated with the same organization.

b) One or more individual(s) in responsible charge of the specification of the vessel and the required design conditions shall certify that the User's Design Specification meets the requirements in paragraph 2.2.2. Such certification requires the signature(s) of one or more Engineers with requisite technical and legal stature, and jurisdictional authority needed for such a document. One or more individuals shall sign the documentation based on information they reviewed, and the knowledge and belief that the objectives of this Division have been satisfied. In addition, these individuals shall prepare a statement to be affixed to the document attesting to compliance with the applicable requirements of the Code (see paragraph 2.A.2.5).

2.A.2.2 Any Engineer that signs and certifies a User's Design Specification shall meet one of the criteria shown below.

a) A Registered Professional Engineer who is registered in one or more of the states of the United States of America or the provinces of Canada and experienced in pressure vessel design.

b) An Engineer experienced in pressure vessel design that meets all required qualifications to perform engineering work and any supplemental requirements stipulated by the user. The Engineer shall identify the location and the licensing or registering authorities under which he has received the authority to perform engineering work.

c) An Engineer experienced in pressure vessel design who meets all required qualifications to perform engineering work and any supplemental requirements stipulated by the user. The Engineer shall be registered in the International Register of Professional Engineers of the Engineers Mobility Forum.

2.A.2.3 The Engineer certifying the User's Design Specification shall comply with the requirements of the location to practice engineering where that Specification is prepared unless the jurisdiction where the vessel will be installed has different certification requirements.

2.A.2.4 When more than one Engineer certifies and signs the User's Design Specification the area of expertise shall be noted next to their signature under "areas of responsibilities" (e.g., design, metallurgy, pressure relief, fabrication, etc.). In addition, one of the Engineers signing the User's Design Specification shall certify that all elements required by this Division are included in the Specification.

2.A.2.5 An example of a typical User's Design Specification Certification Form is shown in Table 2.A.1.

2.A.3 Tables

Table 2.A.1 – Typical Certification Of Compliance Of The User's Design Specification

CERTIFICATION OF COMPLIANCE OF THE USER'S DESIGN SPECIFICATION

I (We), the undersigned, being experienced and competent in the applicable field of design related to pressure vessel requirements relative to this User's Design Specification, certify that to the best of my knowledge and belief it is correct and complete with respect to the Design and Service Conditions given and provides a complete basis for construction in accordance with Part 2, paragraph 2.2.2 and other applicable requirements of the ASME Section VIII, Division 2 Pressure Vessel Code, ______________ Edition with ______________ Addenda and Code Case(s)____________________. This certification is made on behalf of the organization that will operate these vessels______________(company name)_____________

____.

Certified by:____

Title and areas of responsibility: ____

Date: ________________

Certified by:____

Title and areas of responsibility: ____

Date: ________________

Professional Engineer Seal:____(As required)

____

Date: ________________

ANNEX 2.B

GUIDE FOR CERTIFYING A MANUFACTURER'S DESIGN REPORT

(NORMATIVE)

2.B.1 General

An individual(s) in responsible charge of the design and construction of the vessel(s) shall certify that the Manufacturer's Design Report is complete, accurate and in accordance with the User's Design Specification, and that all the requirements of this Division and any additional requirements needed for adequate design have been met. Such certification requires the signature(s) of one or more Engineers with requisite experience and qualifications as defined below. One or more individuals may sign the documentation based on information they reviewed, and the knowledge and belief that the requirements of this Division have been satisfied.

2.B.2 Certification of Manufacturer's Design Report

2.B.2.1 One or a combination of methods shown below shall be used to certify the Manufacturer's Design Report.

a) One or more Professional Engineers, registered in one or more of the states of the United States of America or the provinces of Canada and experienced in pressure vessel design, shall certify the Manufacturer's Design Report meets the requirements in paragraph 2.3.3. The Registered Professional Engineer(s) shall apply the Professional Engineer seal in accordance with the required procedures. In addition, the Registered Professional Engineer(s) shall prepare a statement to be affixed to the document attesting to compliance with the applicable requirements of the Code (see paragraph 2.B.2.6). This Professional Engineer shall be other than the Professional Engineer who certifies the User's Design Specification, although both may be employed by or affiliated with the same organization.

b) One or more individual(s), experienced in pressure vessel design shall certify that the Manufacturer's Design Report meets the requirements in paragraph 2.3.3. Such certification requires the signature(s) of one or more Engineers with requisite technical and legal stature, and corporate authority needed for such a document. These responsible individuals shall sign the documentation based on information they reviewed, and the knowledge and belief that the objectives of this Division have been satisfied. In addition, these individuals shall prepare a statement to be affixed to the document attesting to compliance with the applicable requirements of the Code (see paragraph 2.B.2.6).

2.B.2.2 Any Engineer that signs and certifies a Manufacturer's Design Report shall meet one of the criteria shown below.

a) A Registered Professional Engineer who is registered in one or more of the states of the United States of America or the provinces of Canada and experienced in pressure vessel design.

b) An Engineer experienced in pressure vessel design who meets all required qualifications to perform engineering work and any supplemental requirements stipulated by the user. The Engineer shall identify the location and the licensing or registering authorities under which he has received the authority to perform engineering work stipulated by the user in the Design Specification.

c) An Engineer experienced in pressure vessel design who meets all required qualifications to perform engineering work and any supplemental requirements stipulated by the user. The Engineer shall be registered in the International Register of Professional Engineers of the Engineers Mobility Forum.

2.B.2.3 The Engineer certifying the Manufacturer's Design Report shall comply with the requirements of the location to practice engineering where that Report is prepared unless the jurisdiction where the vessel will be installed has different certification requirements.

2.B.2.4 When more than one Engineer certifies and signs the Manufacturer's Design Report the area of expertise shall be noted next to their signature under "areas of responsibilities" (e.g., design, metallurgy, pressure relief, fabrication, etc.). In addition, one of the Engineers signing the Manufacturer's Design Report shall certify that all elements required by this Division are included in the Report.

2.B.2.5 The inspector shall review the Manufacturer's Design Report and ensure that the requirements of paragraph 2.4.3 have been satisfied.

2.B.2.6 An example of a typical Manufacturer's Design Report Certification Form is shown in Table 2.B.1.

2.B.3 Tables

Table 2.B.1 – Typical Certification Of Compliance Of The Manufacturer's Design Report

CERTIFICATION OF COMPLIANCE OF
THE MANUFACTURER'S DESIGN REPORT

I (We), the undersigned, being experienced and competent in the applicable field of design related to pressure vessel construction relative to the certified User's Design Specification, certify that to the best of my knowledge and belief the Manufacturer's Design Report is complete, accurate and complies with the User's Design Specification and with all the other applicable construction requirements of the ASME Section VIII, Division 2 Pressure Vessel Code, ___________ Edition with _________________Addenda and Code Case(s)_______________________. This certification is made on behalf of the Manufacturer_____________(company name)______
____.

Certified by:_______________________________________
Title and areas of responsibility: ________________________________Date: ________________

Certified by:_______________________________________
Title and areas of responsibility: ________________________________Date: ________________

Professional Engineer Seal:____(As required)
____
Date: ________________

Authorized Inspector Review: ____
Date: _____________

ANNEX 2.C

REPORT FORMS AND MAINTENANCE OF RECORDS

(NORMATIVE)

2.C.1 Manufacturer's Data Reports

2.C.1.1 A Data Report shall be completed by the Manufacturer for each pressure vessel to be marked with the Code symbol.

a) For sample report forms and guidance in preparing Data Reports, see Annex 2.D.

b) A Data Report shall be filled out on Form A-1 or Form A-3 by the Manufacturer and shall be signed by the Manufacturer and the Inspector for each pressure vessel marked with the Code U2 symbol. Same-day production of vessel parts may be reported on a single parts documenting Form A-2 provided all of the following requirements are met:

   1) Vessel Parts are identical

   2) Vessel Parts are manufactured for stock or for the same user or his designated agent

   3) Serial numbers are in uninterrupted sequence

   4) The Manufacturer's written Quality Control System includes procedures to control the development distribution, and retention of the Data Reports

c) The number of lines on the Form A-1 Data Report used to describe multiple components (e.g., nozzles, shell courses) may be increased or decreased as necessary to provide space to describe each component. If addition of lines used to describe multiple components results in the Data Report exceeding one page, space shall be provided for the Manufacturer and Inspector to initial and date each of the additional pages. Horizontal spacing for information on each line may be altered as necessary. All information must be addressed; however, footnotes described in the remarks block are acceptable, e.g., for multiple cases of "none" or "not applicable."

d) Forms may be reprinted, typed, or computer generated.

e) The method of completing the Data Report shall be consistent. The report shall be typed or handwritten using legible printing. Handwritten additions or corrections shall be initialed and dated by the Manufacturer's representative and Inspector.

f) Forms shall not contain advertising slogans, logos, or other commercial matter.

2.C.1.2 Special Requirements For Layered Vessels – A description of the layered shell and/or layered heads shall be given on the Manufacturer's Data Report, describing the number of layers, their thickness or thicknesses, and type of construction (see Table 2.D.2 for the use of Form A-3, Manufacturer's Data Report Supplementary Sheet). An example of the use of Form A-3 illustrating the minimum required data for layered construction is given in Table 2.D.3.

2.C.1.3 The Manufacturer shall distribute the Manufacturer's Data Report as indicated below.

a) Furnish a copy of the Manufacturer's Data Report to the user and, upon request, to the Inspector;

b) Submit a copy of the Manufacturer's Data Report to the appropriate enforcement authority in the jurisdiction in which the vessel is to be installed where required by law;

c) Keep a copy of the Manufacturer's Data Report on file in a safe repository for at least 3 years;

d) In lieu of (b) or (c) above, the vessel may be registered and the Data Reports filed with the National Board of Boiler and Pressure Vessel Inspectors, 1055 Crupper Ave., Columbus, Ohio 43229, USA, where permitted by the jurisdiction in which the vessel is to be installed.

2.C.2 Partial Data Reports

2.C.2.1 The parts Manufacturer shall indicate under “Remarks” the extent the Manufacturer has performed any or all of the design functions. For guidance in preparing Partial Data Reports, see Annex 2D.

2.C.2.2 Partial Data Reports for pressure vessel parts requiring examination under this Division, which are furnished to the Manufacturer responsible for the completed vessel, shall be executed by the parts Manufacturer’s Inspector in accordance with this Division (see paragraph 2.3.1.2). All Partial Data Reports, Form A-2, shall be attached to the Manufacturer’s Data Report, Form A-1.

2.C.3 Maintenance of Records

2.C.3.1 The Manufacturer shall maintain a file for three years after stamping of the vessel, and furnish to the user and, upon request, to the Inspector, the reports and records shown below. It is noted that items that are included in the Manufacturer’s Quality Control System meet the requirements of these subparagraphs.

a) User’s Design Specification (see paragraph 2.2.2)

b) Manufacturer’s Design Report (see paragraph 2.3.3)

c) Manufacturer’s Data Report (see paragraph 2.3.4)

d) Manufacturer’s Construction Records and Partial Data Reports (see paragraph 2.3.5)

   1) Tabulated list of all material used for fabrication with Materials Certifications and Material Test Reports, and a record of any repairs to pressure retaining material that require a radiographic examination by the rules of this Division. The record of the repairs shall include the location of the repair, examination results, and the repair procedures.

   2) Fabrication information including all heat treatment requirements, forming and rolling procedure when prepared, an inspection and test plan identifying all inspection points required by the user, and signed inspection reports

   3) List of any subcontracted services or parts, if applicable

   4) Welding Procedure Specifications (WPS), Procedure Qualification Records (PQR), weld map and welder or welding operator qualification test results

   5) Record of all heat treatments including post weld heat treatment (these records may be either the actual heat treatment charts or a certified summary description of heat treatment time and temperature)

   6) Results of production test plates, if applicable

   7) NDE procedures, records of procedure demonstrations, and records of personnel certifications

   8) All reports stating the results of inspection, nondestructive examinations and testing including radiographic examination, ultrasonic examination, magnetic particle examination, liquid dye penetrant examination and hardness tests

   9) All non-conformance reports including resolution and a detailed description of any repairs including repair procedures, a sketch, photo, or drawing indicating the location and size of the repaired area

   10) Charts or other records of required hydrostatic, pneumatic, or other tests. Test logs shall include the test date, testing fluid, duration of the test, temperature of the test fluid, and test pressure

   11) Dimensional drawings of the as-built condition

2.C.3.2 The Manufacturer shall maintain a complete set of radiographs until the signing of the Manufacturer's Data Report, and furnish upon request to the user and, upon request to the Inspector [see paragraph 7.5.3.1.a)].

ANNEX 2.D

GUIDE FOR PREPARING MANUFACTURER'S DATA REPORTS

(INFORMATIVE)

2.D.1 Introduction

2.D.1.1 The instructions in this Annex provide general guidance to the Manufacturer in preparing the Manufacturer's Data Reports as required in paragraph 2.3.4.

2.D.1.2 Manufacturer's Data Reports required by this Division are not intended for pressure vessels that do not meet the provisions of this Division, including those of special design or construction that require and receive approval by jurisdictional authorities under, laws, rules, and regulations of the respective state or municipality in which the vessel is to be installed.

2.D.1.3 The instructions for completing the Data Reports are identified by numbers corresponding to numbers on the sample forms in this Annex (see Table 2.D.3 forms A-1, A-2 and A-3).

2.D.1.4 Where more space is needed than has been provided on the form for any item, indicate in the space "See Remarks" or "See attached Form A-3," as appropriate.

2.D.1.5 It is not intended that these Data Reports replace in any way the required Manufacturer's Design Report (paragraph 2.3.3) or the Manufacturer's Construction Records (paragraph 2.3.5). It is intended that the Data Reports be used for identifying the vessel, retrieval of records, and certification of compliance with this Division and with the User's Design Specification, by the Manufacturer and by the Inspector.

2.D.2 Tables

Table 2.D.1 – Instructions For The Preparation Of Manufacturer's Data Reports

Applies To Form			Note No.	Instructions
A-1	A-2	A-3		
X	X	X	1	Name and street address of Manufacturer
X	---	X	2	Name and address of purchaser.
X	---	X	3	Name of user, and address where vessel is to be installed.
	X	---	4	Name and address of Manufacturer who will use the vessel part in making the complete vessel
X	X	---	5	Type of vessel, such as horizontal or vertical, tank, separator, heat exchanger, reactor.
	X	---	6	Brief description of vessel part (i.e., shell, two-piece head, tube, bundle).
X	X	X	7	An identifying Manufacturer's serial number marked on the vessel (or vessel part) (see Annex 2-F).
X	X	X	8	Applicable Jurisdiction Registration No.
X	X	---	9	Indicate drawing numbers, including revision numbers, which cover general assembly and list materials. For Canadian registration, the number of the drawing approved by the applicable jurisdictional authority.
	X	---	10	Organization that prepared drawing.
X	X	X	11	Where applicable, National Board Number from Manufacturer's Series of National Board Numbers. National Board Number shall not be used for owner-inspected vessels
X	X	---	12	Issue date of Section VIII, Division 2 and Addenda under which vessel was manufactured
X	X	---	13	All code case numbers when the vessel is manufactured to any Code Cases.
X	---	---	14	To be completed when one or more parts of the vessel are furnished by others and certified on Data Report Form A-2 as required by Annex 2-F. The part manufacturer's name and serial number should be indicated.
X	X	---	15	Show the complete ASME Specification number and grade of the actual material used in the vessel part. Material is to be as designated in Section VIII, Division 2 (e.g., "SA-285 C"). Exceptions: A specification number for a material not identical to an ASME Specification may be shown only if such material meets the criteria in the Foreword of this Section. When material is accepted through a Code Case, the applicable Case Number shall be shown.
X	X	---	16	Thickness is the nominal thickness of the material used in the fabrication of the vessel. It includes corrosion allowance.
X	X	---	17	State corrosion allowance on thickness.
X	X	---	18	Indicate whether the diameter is inside diameter or outside diameter.
X	X	---	19	The shell length shall be shown as the overall length between closure or transition section welds, for a shell of a single diameter. In other cases, define length, as appropriate.
X	X	---	20	Type of longitudinal joint in cylindrical section, or any joint in a sphere (e.g., Type No.1 butt, or seamless) per Part 4, paragraph 4.2.
X	X	---	21	State the temperature and time if heat treatment is performed by the Manufacturer (i.e. postweld heat treatment, annealing, or normalizing). Explain any special cooling procedure under "Remarks."
X	X	---	22	Indicate examination applied to longitudinal seams. Any additional examinations should be included under "Remarks."
X	X	---	23	Type of welding used in girth joints in the cylindrical section (see 20).
X	X	---	24	Indicate examination applied to girth joints (see 22).
X	X	---	25	Number of cylindrical courses, or belts, required to make one shell.
X	X	---	26	Show specified minimum thickness of head after forming. State dimensions that define the head shape.
X	X	---	27	Bolts used to secure removable head or heads of vessel.
X	X	---	28	For jacketed vessels, explain the type of jacket closures used.
X	X	---	29	Show the internal maximum allowable working pressure and the external maximum allowable working pressure.
X	X	---	30	Show the coincident temperatures that correspond to the internal maximum allowable working pressure and the external maximum allowable working pressure, as applicable.
X	X	---	31	Show minimum Charpy V-notch impact value required and impact test temperature. If exempted, indicate under "Remarks" paragraph under which exemption was taken.
X	X	---	32	Show minimum design metal temperature

Table 2.D.1 – Instructions For The Preparation Of Manufacturer's Data Reports

Applies To Form			Note No.	Instructions
A-1	A-2	A-3		
X	X	---	33	Show hydrostatic or other tests made with specified test pressure at top of vessel in the test position. Cross out words (pneumatic, hydrostatic, or combination test pressure) that do not apply. Indicate under "Remarks" if vessel was tested in the vertical position. See Part 8 for special requirements for combination units
X	X	---	34	Indicate nozzle or other opening that is designated for pressure relief.
X	X	---	35	Show other nozzles and openings by size and type (see 50)
X	X	---	36	Show opening designated for inspection. Show location.
X	X	---	37	Indicate provisions for support of the vessel and any attachments for superimposed equipment.
X	X	---	38	Indicate whether fatigue analysis is required per Part 4.
X	X	---	39	Describe contents or service of the vessel.
X	X	---	40	Space for additional comments, including any Code restrictions on the vessel or any unusual Code requirements that have been met, such as those noted in 21, 22, 24, 31, and 33, or in Part 1, paragraphs 1.2.1 and 1.2.2, or Part 5, paragraph 5.10. Indicate stiffening rings, if used.
X	X	---	41	Certificate of compliance block is to show the name of the Manufacturer as shown on his ASME Code Certificate of Authorization. This should be signed in accordance with organizational authority defined in the Quality Control System (see Annex 2.E).
X	X	---	42	This certificate is to be completed by the Manufacturer to show the disposition of the User's Design Specification and the Manufacturer's Design Report, and to identify the individuals who certify them per Part 2, paragraphs 2.2.2 and 2.3.2, respectively (see 49).
X	X	X	43	This certificate is to be completed by the Manufacturer and signed by the Authorized Inspector who performs the shop inspection.
X	X	X	44	This Inspector's National Board Commission Number must be shown when the vessel is stamped "National Board." Otherwise, show only his state or province Commission Number.
X	---	---	45	This certificate is for the Authorized Inspector to sign for any field construction or assembly work (see 44 for National Board Commission Number requirements). Indicate the method used to pressure test the vessel.
---	---	X	46	Fill in information identical to that shown on the Data Report to which this sheet is supplementary.
---	---	X	47	Fill in information for which there was insufficient space for a specific item on the Data Report Form as identified by the notation "See attached Form A-3" on the Data Report. Identify the information by the applicable Data Report Item Number.
---	X	---	48	Indicate data, if known.
X	X	---	49	Registration Locale.
X	X	---	50	Data entries with descriptions acceptable to Inspector. Abbreviations, coded identification, or reference to Code Figure and sketch number may be used to define any generic name. For ASME B16.5 flanges, the class should be identified. Flange facing and attachment to neck is not required. Some typical abbreviations are shown below • Flanged fabricated nozzle: Cl. 300 flg • Long weld neck flange: Cl. 300 lwn • Weld end fabricated nozzle: w.e.
X	X	---	51	Material for nozzle neck. Flange material not necessary.
X	X	---	52	Nominal nozzle neck thickness. For ASME B16.ll and similar parts, class designation may be substituted for thickness.

Table 2.D.2 – Supplementary Instructions For The Preparation Of Manufacturer's Data Reports For Layered Vessels

Note Letter	Instructions
A	Letter symbols indicate instructions that supplement the instructions of Table 2.D.1
B	The Form A3L is not available preprinted as shown. It is intended as an example of suggested use of Form A-3 for reporting data for a vessel of layered construction. It is intended that the Manufacturer develop his own arrangement to provide supplementary data that describes his vessel.
C	Note the NDE performed (RT, PT, MT, UT).
D	Applies only when heads are of layered construction.
E	Indicates if seamless or welded.
F	When more than one layer thickness is used, add lines as needed.
G	Indicate diameter of vent holes in the layers.
H	Indicate whether vent holes are in random locations in each layer, or are drilled through all layers.
I	Indicate locations of nozzles and openings; layered shell; layered head.
J	Indicate method of attachment and reinforcement of nozzles and openings in layered shells and layered heads. Refer to figure number if applicable.

Table 2.D.3 – Manufacturer's Data Report Forms

FORM A-1 MANUFACTURER'S DATA REPORT FOR PRESSURE VESSELS
As Required by the Provisions of the ASME Code Rules, Section VIII, Division 2

1. Manufactured and certified by 1
(Name and address of manufacturer)

2. Manufactured for 2
(Name and address of purchaser)

3. Location of installation 3
(Name and address)

4. Type 5 (Horiz. or vert. tank) 7 (Mfr.'s Serial No.) 8 (CRN) 9 (Drawing No.) 11 (Nat'l. Bd. No.) (Year built)

5. The chemical and physical properties of all parts meet the requirements of material specifications of the ASME BOILER AND PRESSURE VESSEL CODE. The design, construction, and workmanship conform to ASME Code, Section VIII, Division 2.
12 (Year) 12 (Addenda date) 13 (Code case No.)

Items 6 to 11 incl. to be completed for single wall vessels, jackets of jacketed vessels, or shells of heat exchangers

6. Shell 15 (Material (Spec. No., Grade)) 16 (Nom. thk.) 17 (Corr. allow.) 18 (diameter) 19 (Length (overall))

7. Seams 20 (Longitudinal) 21 (Heat treatment) 22 (Nondestructive Examination)
23 (Girth) 21 (Heat treatment) 24 (Nondestructive Examination) 25 (No. of Courses)

8. Heads: (a) Matl. 15,20,21,22 (Spec., No., Grade) (b) Matl. 15,20,21,22 (Spec., No., Grade)

	Location (Top, Bottom, End)	Minimum Thickness	Corrosion Allowance	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a)		26	17							
(b)										

9. If removable, bolts used (describe other fastenings): 27
Matl. Spec. No. Grade Size Number

10. Jacket closure 28 (Describe as ogee and weld, bar. etc) If bar, give dimensions ______ If bolted, describe or sketch.

11. MAWP 29 (internal) 29 (external) at max. temp. 30 (internal) 30 (external) Min. design metal temp. 32 at 32
Impact test 31 At test temperature of 31
Hydro., pneu., or comb test pressure 33

Items 12 and 13 to be completed for tube sections

12. Tubesheets 15 (Stationary matl. (Spec. No., Grade)) (Diam. (Subject to pressure)) 16 (Nom. thk.) 17 (Corr. Allow.) (Attach. (wld., bolted))
15 (Floating matl. (Spec. No., Grade)) (Diam.) 16 (Nom. thk.) 17 (Corr. Allow.) (Attach. (wld., bolted))

13. Tubes 15 (Matl. (Spec. No.. Grade)) (O.D.) (Nom. thk.) (Number) (Type (straight or "U"))

Items 14 to 18 incl. to be completed for inner chambers of jacketed vessels, or channels of heat exchangers

14. Shell 15 (Material (Spec. No., Grade)) 16 (Nom. thk.) 17 (Corr. allow.) (diameter) 19 (Length (overall))

15. Seams 20 (Longitudinal) 21 (Heat treatment) 22 (Nondestructive Examination)
23 (Girth) 21 (Heat treatment) 24 (Nondestructive Examination) (No. of Courses)

16. Heads: (a) Matl. (Spec., No., Grade) (b) Matl. (Spec., No., Grade)

	Location (Top, Bottom, End)	Minimum Thickness	Corrosion Allowance	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a)										
(b)										

17. If removable, bolts used (describe other fastenings):
Matl. Spec. No. Grade Size Number

18. MAWP 29 (internal) 29 (external) at max. temp. 30 (internal) 30 (external) Min. design metal temp. 32 at 32
Impact test 31 At test temperature of 31
Hydro., pneu., or comb test pressure 33

This Form (EOO1121) may be obtained from ASME, Order Dept.. 22 Law Drive. Box 2300, Fairfield. NJ 07007-2300

2007 SECTION VIII — DIVISION 2

FORM A-1 (Back)

Items below to be completed for all vessels where applicable

19. Nozzles inspection and safety valve openings

Purpose (Inlet, Outlet, Drain, etc)	No	Diam. or Size	Type	Material	Nom. Thk.	Reinforcement Material	How Attached	Location
34,35,36		35	35,50	15,51	16,52			36

20. Support Skirt 37 (Yes or No) Lugs ______ (No.) Legs ______ (No) Other ______ (Describe) Attached ______ (Where and how)

21. Service: Fatigue analysis required 38 (Yes or No) and 39 (Describe contents or service)

Remarks: 21,22,24,31,33,37,40,47

42

CERTIFICATION OF DESIGN

Users Design Specification on file at ______

Manufacturer's Design Report on file at ______

User's Design Specification certified by ______ PE State 42,49 Reg. No. ______

Manufacturer's Design Report certified by ______ PE State 42,49 Reg. No. ______

41

CERTIFICATE OF SHOP COMPLIANCE

We certify that the statements in this report are correct and that all details of design, material, construction, and workmanship of this vessel conforms to the ASME Code for Pressure Vessels, Section VIII, Division 2.

"U2" Certificate of Authorization No 41 expires ______

Date ______ Co. name 41 (Manufacturer) Signed 41 (Representative)

43

CERTIFICATE OF SHOP INSPECTION

Vessel made by ______ at ______

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State or Province of ______ and employed by ______ of ______

, have inspected the pressure vessel described in this Manufacturer's Data Report on ______

, and state that, to the best of my knowledge and belief, the Manufacturer has constructed this pressure vessel in accordance with ASME Code, Section VIII, Division 2. By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this Manufacturer's' Data Report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date ______ Signed 43 (Authorized Inspector) Commissions 41 (Nat'l. Board (incl. Endorsements), State, Prov. and No.)

41

CERTIFICATE OF FIELD ASSEMBLY COMPLIANCE

We certify that the field assembly construction of all parts of this vessel conforms with the requirements of SECTION VIII, Division 2 of the ASME BOILER AND PRESSURE VESSEL CODE.

"U2" Certificate of Authorization No ______ expires ______

Date ______ Co. name ______ (Assembler that certified and constructed field assembly) Signed ______ (Representative)

45

CERTIFICATE OF FIELD ASSEMBLY INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State or Province of ______ and employed by ______ of ______

have compared the statements in this Manufacturer's Data Report with the described pressure vessel and state that parts referred to as data items ______

not included in the certificate of shop inspection, have been inspected by me and that, to the best of my knowledge and belief, the Manufacturer has constructed and assembled this pressure vessel in accordance with the ASME Code, Section VIII, Division 2

The described vessel was inspected and subjected to a hydrostatic test of. ______

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this Manufacturer's Data Report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date ______ Signed ______ (Authorized Inspector) Commissions ______ (Nat'l. Board (incl. Endorsements), State, Prov. and No.)

2010 SECTION VIII, DIVISION 2

FORM A-2 MANUFACTURER'S PARTIAL DATA REPORT

A PART OF A pressure Vessel Fabricated by One Manufacturer for Another Manufacturer

As Required by the Provisions of the ASME Code Rules, Section VIII, Division 2

1. Manufactured and certified by 1
   (Name and address of manufacturer)

2. Manufactured for 4
   (Name and address of purchaser)

3. Location of installation 3
   (Name and address)

4. Type

5	7	8	9	11	
Horiz. or vert. tank	Mfr.'s Serial No.	CRN	Drawing No.	Nat'l. Bd. No.	Year built

5. The chemical and physical properties of all parts meet the requirements of material specifications of the ASME BOILER AND PRESSURE VESSEL CODE. The design, construction, and workmanship conform to ASME Code, Section VIII, Division 2.

12	12	13
Year	Addenda date	Code case No.

6. Constructed to:

		6
Drawing No.	Drawing Prepared by	Description of part inspected

Items 7 to 12 incl. to be completed for single wall vessels, jackets of jacketed vessels, or shells of heat exchangers

7. Shell

15	16	17	18	19
Material (Spec. No., Grade)	Nom. thk.	Corr. allow.	diameter	Length (overall)

8. Seams

20	21	22
Longitudinal	Heat treatment	Nondestructive Examination

23	21	24	25
Girth	Heat treatment	Nondestructive Examination	No. of Courses

9. Heads: (a) Matl. 15,20,21,22 (Spec., No., Grade) (b) Matl. 15,20,21,22 (Spec., No., Grade)

	Location (Top, Bottom, End)	Minimum Thickness	Corrosion Allowance	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a)		26	17							
(b)										

10. If removable, bolts used (describe other fastenings): 27
    Matl. Spec. No. Grade Size Number

11. Jacket closure 28 If bar, give dimensions ______ If bolted, describe or sketch.
    Describe as ogee and weld, bar. etc

12. MAWP 29 (internal) 29 (external) at max. temp. 30 (internal) 30 (external) Min. design metal temp. 32 at 32
    Impact test 31 At test temperature of 31
    Hydro., pneu., or comb test pressure 33

Items 13 and 14 to be completed for tube sections.

13. Tubesheets

15	18	16	17	
Stationary matl. (Spec. No., Grade)	Diam. (Subject to pressure)	Nom. thk.	Corr. Allow.	Attach. (wld., bolted)
15		16	17	
Floating matl. (Spec. No., Grade)	(Diam.)	Nom. thk.	Corr. Allow.	Attach. (wld., bolted)

14. Tubes

15				
Matl. (Spec. No.. Grade)	***O.D.***	***Nom. thk.***	***Number***	***Type (straight or "U")***

Items 15 to 18 incl. to be completed for inner chambers of jacketed vessels, or channels of heat exchangers

15. Shell

15	16	17	18	19
Material (Spec. No., Grade)	Nom. thk.	Corr. allow.	diameter	Length (overall)

16. Seams

20	21	22
Longitudinal	Heat treatment	Nondestructive Examination

23	21	24	
Girth	Heat treatment	Nondestructive Examination	No. of Courses

17. Heads: (a) Matl. ______ (Spec., No., Grade) (b) Matl. ______ (Spec., No., Grade)

	Location (Top, Bottom, End)	Minimum Thickness	Corrosion Allowance	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a)										
(b)										

18. If removable, bolts used (describe other fastenings): ______
    Matl. Spec. No. Grade Size Number

19. Design press. 29 at max. temp. 30 . Charpy impact 31
    at test temp. of 31 . Min. design metal temp. 32 at ______
    Pneu., hydro., or comb. pressure test 33

This Form (EOO114) may be obtained from ASME, Order Dept.. 22 Law Drive. Box 2300, Fairfield. NJ 07007-2300.

FORM A-2 (Back)

Items below to be completed for all vessels where applicable

20. Nozzles inspection and safety valve openings

Purpose (Inlet, Outlet, Drain, etc)	No	Diam. or Size	Type	Material	Nom. Thk.	Reinforcement Material	How Attached	Location
34		35	35	15	16			36
35			50	51	52			
36								
48								

21. Support Skirt 37 (Yes or No) Lugs ______ (No.) Legs ______ (No) Other ______ (Describe) Attached ______ (Where and how)

Remarks: 21,22,24,31,33,37,40,47

42

CERTIFICATION OF DESIGN

Users Design Specification on file at ______

Manufacturer's Design Report on file at ______

User's Design Specification certified by ______ PE State 42,49 Reg. No. ______

Manufacturer's Design Report certified by ______ PE State 42,49 Reg. No. ______

41

CERTIFICATE OF SHOP COMPLIANCE

We certify that the statements in this report are correct and that all details of design, material, construction, and workmanship of this vessel conforms to the ASME Code for Pressure Vessels, Section VIII, Division 2.

"U2" Certificate of Authorization No 41 expires ______

Date ______ Co. name 41 (Manufacturer) Signed 41 (Representative)

43

CERTIFICATE OF SHOP INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State or Province of ______ and employed by ______ of ______, have inspected the part of a pressure vessel described in this Manufacturer's Data Report on ______, and state that, to the best of my knowledge and belief, the Manufacturer has constructed this part in accordance with ASME Code, Section VIII, Division 2. By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the part described in this Manufacturer's' Data Report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date ______ Signed 43 (Authorized Inspector) Commissions 41 (Nat'l. Board (incl. Endorsements), State, Prov. and No.)

FORM A-3 MANUFACTURER'S DATA REPORT SUPPLEMENTARY SHEET
As Required by the Provisions of the ASME Code Rules, Section VIII, Division 2

1. Manufactured and certified by ______ 1 ______
(Name and address of manufacturer)

2. Manufactured for ______ 2 ______
(Name and address of purchaser)

3. Location of installation ______ 3 ______
(Name and address)

4. Type

5	7	8	9	11	
Horiz. or vert. tank	Mfr.'s Serial No.	CRN	Drawing No.	Nat'l.. Bd. No.	Year built

Data Report Item Number	Remarks
46	47

Date		Co. name	43,46 Manufacturer	Signed	43,46 Representative
Date		Signed	43,46 Authorized Inspector	Commissions	43,46 Nat'l. Board (incl. Endorsements), State, Prov. and No.

This form (E00119) may be obtained from the ASME, Order Dept., 22 Law Drive, Box 2300, Fairfield, NJ 07007-2300.

FORM A-3 MANUFACTURER'S DATA REPORT SUPPLEMENTARY SHEET

As Required by the Provisions of the ASME Code Rules, Section VIII, Division 2

1. Manufactured and certified by ______ 1 ______
(Name and address of manufacturer)

2. Manufactured for ______ 2 ______
(Name and address of purchaser)

3. Location of installation ______ 3 ______
(Name and address)

4. Type

5	7	8	9	11	
Horiz. or vert. tank	Mfr.'s Serial No.	CRN	Drawing No.	Nat'l. Bd. No.	Year built

Data Report 46 Item Number	A,B,47 Remarks					
Item 6 or 7	(a) layered construction type: (Concentric, wrapped, spiral, coil wound, shrink fit, etc.) Nom. Layer					
(shell)	Location	Mat'l.	Layer Thk.	Nom. Thk.Tot.	No. Courses	NDE
	(b) Inner Shell					
	(c) Dummy Layer	15	F	16	25	c
	(d) Layers:					
	(e) Overwraps:					
Item 8	(a) Layered Construction Type: *{Formed. Machined.. Segmental, etc.)*					
(heads)	(b) Inner Head					
	(c) Dummy Layer	15	F	16	E,20	c
	(d) Layers:					
	(a) Layered Construction Type:					
	(1) Inner Head					
	(2) Dummy Layer					
	(3) Layers:					
Item 21		Diam Hole	Staggered Layers Or Radial Through			
(Vent Holes in Layers)	(a) Layered Shell		H			
	(b) Layered Head	G				
Item 24	Gaps Have Been Controlled According to the Provisions of Paragraph:					
(Remarks)	(See 4.13.12.1, 14.13.12.2 and 14.13.12.3)					
	I,J					
	B					

Date ______ Co. name ______ Signed ______
Manufacturer Representative

Date ______ Signed ______ Commissions ______
Authorized Inspector Nat'l. Board (incl. Endorsements), State, Prov. and No.

This form (E00119) may be obtained from the ASME, Order Dept., 22 Law Drive, Box 2300, Fairfield, NJ 07007-2300.

ANNEX 2.E

QUALITY CONTROL SYSTEM

(NORMATIVE)

2.E.1 General

2.E.1.1 The Manufacturer shall have and maintain a Quality Control System that will establish that all Code requirements, including material, design, fabrication, examination (by the Manufacturer), and inspection of vessels and vessel parts (by the Inspector), will be met. Provided that Code requirements are suitably identified, the system may include provisions for satisfying any requirements by the Manufacturer or user that exceed minimum Code requirements and may include provisions for quality control of non-Code work. In such systems, the Manufacturer of vessels and vessel parts may make changes in parts of the system that do not affect the Code requirements without securing acceptance by the Inspector (see paragraph 2.1.1). When revisions are made to Quality Control Systems of Manufacturers of pressure relief valves, they must be accepted by the ASME designated organization before implementation if such revisions affect Code requirements.

2.E.1.2 The system that the Manufacturer uses to meet the requirements of this Division shall be one suitable for the Manufacturer's circumstances. The necessary scope and detail of the system shall depend on the complexity of the work performed and on the size and complexity of the Manufacturer's organization. A written description of the system the Manufacturer will use to produce a Code item shall be available for review. Depending upon the circumstances, the description may be brief or extensive.

2.E.1.3 The written description may contain information of a proprietary nature relating to the Manufacturer's processes. Therefore, the Code does not require any distribution of this information except for the Inspector's or ASME designee's copy as covered by paragraph 2.E.15.3 and 2.E.16.3. It is intended that information learned about the system in connection with the evaluation will be treated as confidential and that all loaned descriptions will be returned to the Manufacturer upon completion of the evaluation.

2.E.1.4 The Quality Control System of UV Stamp Holders shall be in accordance with the requirements of Division 1.

2.E.2 Outline of Features Included in the Quality Control System

The following is a guide to some of the features which should be covered in the written description of the Quality Control System and is equally applicable to both shop and field work.

a) The information associated with paragraph 2.3 and Annex 7A.

b) The complexity of the work includes factors such as design simplicity versus complexity, the types of materials and welding procedures used, the thickness of materials, the types of nondestructive examinations applied, and whether heat treatments are applied.

c) The size and complexity of the Manufacturer's organization includes factors such as the number of employees, the experience level of employees, the number of vessels produced, and whether the factors defining the complexity of the work cover a wide or narrow range.

2.E.3 Authority and Responsibility

The authority and responsibility of those in charge of the Quality Control System shall be clearly established. Persons performing quality control functions shall have sufficient and well-defined responsibility, the authority, and the organizational freedom to identify quality control problems and to initiate, recommend, and provide solutions.

2.E.4 Organization

An organization chart showing the relationship between management and engineering, purchasing, manufacturing, field construction, inspection, and quality control is required to reflect the actual organization. The purpose of this chart is to identify and associate the various organizational groups with the particular function for which they are responsible. The Code does not intend to encroach on the Manufacturer's right to establish, and from time to time to alter, whatever form of organization the Manufacturer considers appropriate for its Code work.

2.E.5 Drawings, Design Calculations, and Specification Control

The Manufacturer's Quality Control System shall provide procedures which will ensure that the latest applicable drawings, design calculations, specifications, and instructions, required by the Code, as well as authorized changes, are used for manufacture, assembly, examination, inspection, and testing. The system shall ensure that authorized changes are included, when appropriate, in the User's Design Specification and/or in the Manufacturer's Design Report.

2.E.6 Material Control

The Manufacturer shall include a system of receiving control that will ensure that the material received is properly identified and has documentation including required material certifications or material test reports to satisfy Code requirements as ordered. The system material control shall ensure that only the intended material is used in Code construction.

2.E.7 Examination and Inspection Program

The Manufacturer's Quality Control System shall describe the fabrication operations, including examination, sufficiently to permit the Inspector or ASME designee to determine at what stages specific inspections are to be performed.

2.E.8 Correction of Nonconformities

There shall be a system agreed upon with the Inspector for correction of nonconformities. A nonconformity is any condition which does not comply with the applicable rules of this Division. Nonconformities must be corrected or eliminated in some way before the completed component can be considered to comply with this Division.

2.E.9 Welding

The Quality Control System shall include provisions for indicating that welding conforms to requirements of Section IX as supplemented by this Division.

2.E.10 Nondestructive Examination

The Quality Control System shall include provisions for identifying nondestructive examination procedures the Manufacturer or Assembler will apply to conform to the requirements of this Division.

2.E.11 Heat Treatment

The Quality Control System shall provide controls to ensure that heat treatments as required by the rules of this Division are applied. Means shall be indicated by which the Inspector or ASME designee will be ensured that these Code heat treatment requirements are met. This may be by review of furnace time-temperature records or by other methods as appropriate.

2.E.12 Calibration of Measurement and Test Equipment

The Manufacturer shall have a system for the calibration of examination, measuring, and test equipment used in fulfillment of requirements of this Division.

2.E.13 Records Retention

The Manufacturer shall have a system for the maintenance of Data Reports and records as required by this Division. Requirements for maintenance of records are given in paragraph 2.C.3. Additionally, retained records as required by this Division and the Quality Control System shall be made available to the Authorized Inspector Supervisors or to review teams designated by ASME.

2.E.14 Sample Forms

The forms used in this Quality Control System and any detailed procedures for their use shall be available for review. The written description shall make necessary references to these forms.

2.E.15 Inspection of Vessels and Vessel Parts

2.E.15.1 Inspection of vessels and vessel parts shall be by the Inspector as defined in paragraph 2.4.

2.E.15.2 The written description of the Quality Control System shall include reference to the Inspector.

2.E.15.3 The Manufacturer shall make available to the Inspector, at the Manufacturer's plant or construction site, a current copy of the written description of the Quality Control System.

2.E.15.4 The Manufacturer's Quality Control System shall provide for the Inspector at the Manufacturer's plant to have access to the User's Design Specification, the Manufacturer's Design Report, and all drawings, calculations, specifications, procedures, process sheets, repair procedures, records, test results, and other documents as necessary for the Inspector to perform his duties in accordance with this Division. The Manufacturer may provide such access either to his own files of such documents or by providing copies to the Inspector.

2.E.16 Inspection of Pressure Relief Valves

2.E.16.1 Inspection of pressure relief valves shall be by a designated representative of ASME, as described in Part 9.

2.E.16.2 The written description of the Quality Control System shall include reference to the ASME designee.

2.E.16.3 The valve Manufacturer shall make available to the ASME designee, at the Manufacturer's plant, a current copy of the written description of the applicable Quality Control System.

2.E.16.4 The valve Manufacturer's Quality Control System shall provide for the ASME designee to have access to all drawings, calculations, specifications, procedures, process sheets, repair procedures, records,

test results, and any other documents as necessary for the designee to perform his duties in accordance with this Division. The Manufacturer may provide such access either to his own files of such documents or by providing copies to the designee.

ANNEX 2.F

CONTENTS AND METHOD OF STAMPING

(NORMATIVE)

2.F.1 Required Marking for Vessels

Each pressure vessel to which the U2 symbol is applied shall be marked with the following.

a) The U2 symbol, as shown in Figures 2.F.1, shall be stamped on vessels certified in accordance with this Division.

b) The name of the Manufacturer of the pressure vessel as it is shown on the Certificate of Authorization or an abbreviation accepted by ASME, preceded by "Certified by." A trademark is not considered to be sufficient identification for vessels or parts constructed to this Division.

c) The Manufacturer's serial number (MFG SER).

d) The MAWP (Maximum Allowable Working Pressure) at the coincident maximum design metal temperature. When a vessel is specified to operate at more than one pressure and temperature condition, such values of coincident pressure and design temperature shall be added to the required markings.

e) The MDMT (minimum design metal temperature) at coincident MAWP in accordance with Part 3.

f) The year built.

g) Code Edition and Addenda (see 2.1.3)

h) The construction type, all of the applicable construction types shall be included.

   1) F – Forged

   2) W – Welded

   3) WL – Welded layered

i) Heat treatment markings shall be as follows:

   1) The letters HT shall be applied under the Code Symbol when the complete vessel has been post weld heat treated in accordance with Part 3.

   2) The letters PHT shall be applied under the Code Symbol when only part of the complete vessel has been post weld heat treated in accordance with Part 3.

j) When inspected by a user's Inspector as provided in paragraph 2.4.1, the word USER shall be marked above the Code Symbol.

2.F.2 Methods of Marking Vessels with Two or More Independent Chambers

One of the following arrangements shall be used in marking vessels having two or more independent pressure chambers designed for the same or different operating conditions. Each detachable chamber shall be marked to identify it positively with the combined unit.

a) If markings are grouped in one location, then the markings may be grouped in one location on the vessel provided they are arranged to indicate clearly the data applicable to each chamber, including the maximum differential pressure for the common elements, when this pressure is less than the higher pressure in the adjacent chambers.

b) If each independent chamber is marked, then the complete required marking may be applied to each independent pressure chamber, provided additional marking, such as name of principal chamber (e.g., process chamber, jacket, tubes) is used to indicate clearly to which chamber the data apply.

2.F.3 Application of Stamp

The U2 symbol shall be applied by the Manufacturer only with the approval of the Inspector, and after the hydrostatic test and all other required inspection and testing has been satisfactorily completed. Such application of the ASME U2 symbol, together with final certification in accordance with the rules of this Division, shall confirm that all applicable requirements of this Division and the User's Design Specification have been satisfied.

2.F.4 Part Marking

2.F.4.1 Parts of pressure vessels for which Partial Data Reports are required shall be marked by the parts Manufacturer with the following:

a) The appropriate ASME symbol stamp shown in Figure 2.F.1 above the word “PART”.

b) The name of the Manufacturer of the part, preceded by the words “Certified by”.

c) The Manufacturer's serial number assigned to the part.

d) The MAWP and coincident maximum design metal temperature (see Part 2).

e) The MDMT (minimum design metal temperature) at the MAWP (see Part 3).

2.F.4.2 The requirements for part marking in accordance with paragraph 2.F.4.1.d and 2.F.4.1.e do not apply for overpressure relief devices that are covered in Part 9.

2.F.5 Application of Markings

Markings required in paragraphs 2.F.1 through 2.F.4 shall be applied by one of the following methods

a) Nameplate – A separate metal nameplate, of a metal suitable for the intended service, at least 0.5 mm (0.02 in.) thick, shall be permanently attached to the vessel or to a bracket that is permanently attached to the vessel. The nameplate and attachment shall be such that removal shall require willful destruction of the nameplate or its attachment system. The attachment weld to the vessel shall not adversely affect the integrity of the vessel. Attachment by welding shall not be permitted on materials enhanced by heat treatment or on vessels that have been pre-stressed.

   1) Only the U2 symbol need be stamped on the nameplate.

   2) All other data may be stamped, etched, or engraved on the nameplate (see paragraph 2.F.7).

b) Directly on Vessel Shell – Markings shall be stamped, with low stress type stamps, directly on the vessel, located on an area designated as a low stress area by the Manufacturer in the Manufacturer's Design Report (see paragraph 2.3.3).

c) Adhesive Attachment – Nameplates may be attached with pressure-sensitive acrylic adhesive systems in accordance with the following requirements.

   1) Adhesive systems for the attachment of nameplates are limited to:

      i) The use of pressure-sensitive acrylic adhesives that have been pre-applied by the nameplate manufacturer to a nominal thickness of at least 0.13 mm (0.005 in.) and that are protected with a moisture-stable liner

      ii) Use for vessels with design temperatures within the range of -40°C to 150°C (-40°F to 300°F) inclusive

iii) Application to clean, bare metal surfaces, with attention being given to removal of anti-weld spatter compound that may contain silicone

iv) Use of pre-qualified application procedures as outlined in paragraph 2.F.5.c.2

v) Use of the pre-applied adhesive within an interval of 2 years after adhesive application

2) Nameplate Application Procedure Qualification

i) The Manufacturer's Quality Control System (see Annex 2.E) shall define that written procedures, acceptable to the Inspector, for the application of adhesive-backed nameplates shall be prepared and qualified.

ii) The application procedure qualification shall include the following essential variables, using the adhesive and nameplate manufacturers' recommendations where applicable:

- Description of the pressure-sensitive acrylic adhesive system employed, including generic composition
- The qualified temperature range, the cold box test temperature shall be -40°C (-40°F) for all applications
- Materials of nameplate and substrate when the mean coefficient of expansion at design temperature of one material is less than 85% of that for the other material
- Finish of the nameplate and substrate surfaces
- The nominal thickness and modulus of elasticity at application temperature of the nameplate when nameplate pre-forming is employed – a change of more than 25% in the quantity: [(nameplate nominal thickness)^2 x nameplate modulus of elasticity at application temperature] will require re-qualification
- The qualified range of preformed nameplate and companion substrate contour combinations when pre-forming is employed
- Cleaning requirements for the substrate
- Application temperature range and application pressure technique
- Application steps and safeguards

iii) Each procedure used for nameplate attachment by pressure-sensitive acrylic adhesive systems shall be qualified for outdoor exposure in accordance with Standard UL-969, Marking and Labeling Systems, with the following additional requirements.

- Width of nameplate test strip shall not be less than 25 mm (1 in.).
- Nameplates shall have an average adhesion of not less than 1.4 N/mm (8 lb/in.) of width after all exposure conditions, including low temperature.
- Any change in paragraph 2.F.5.c.2.ii shall require re-qualification.
- Each lot or package of nameplates shall be identified with the adhesive application date.

2.F.6 Duplicate Nameplate

A duplicate nameplate may be attached on the support, jacket, or other permanent attachment to the vessel. All data on the duplicate nameplate, including the U2 symbol, shall be cast, etched, engraved, or stamped. The Inspector need not witness this marking. The duplicate nameplate shall be marked "DUPLICATE." The use of duplicate nameplates, and the stamping of the ASME symbol on the duplicate nameplate, shall be controlled as described in the Manufacturer's Quality Control System.

2.F.7 Size and Arrangements of Characters for Nameplate and Direct Stamping of Vessels

2.F.7.1 The data shall be in characters not less than 8 mm (5/16 in.) high and shall be arranged substantially as shown in Figures 2.F.1. Characters shall be either indented or raised at least 0.10 mm (0.004 in. and shall be legible and readable.

2.F.7.2 Where space limitations do not permit the requirements of paragraph 2.F.7.1 to be met, such as for parts with outside diameters of 89 mm (3.5 in.) or smaller, the required character size to be stamped directly on the vessel may be 3 mm (1/8 in.).

2.F.8 Attachment of Nameplate or Tag

If all or part of the data is marked on the nameplate or tag before it is attached to the vessel, the Manufacturer shall ensure that the nameplate with the correct marking has been attached to the vessel to which it applies as described in their Quality Control System. The Inspector shall verify that this has been done.

2.F.9 Figures

U2 Letters Denoting The Construction Type (see paragraph 2.F.1.g)	Certified by ____________________ (Name of Manufacturer) __________ at __________ Maximum Allowable Working Pressure __________ at __________ Maximum Allowable External Working Pressure (Note 2) __________ at __________ Minimum Design Metal Temperature ____________________ Manufacturer's Serial Number ____________________ Year Built ____________________ Code Edition & Addenda

Notes:

1) Information within parentheses is not part of the required marking. Phrases identifying data may be abbreviated; minimum abbreviations shall be MAWP, MDMT, S/N, and year, respectively.
2) The maximum allowable external working pressures required only when specified as a design condition.

Figure 2.F.1 – Form of Stamping

ANNEX 2.G

OBTAINING AND USING CODE STAMPS

(NORMATIVE)

2.G.1 Code Stamps Bearing Official Symbol

A Certificate of Authorization to use the Code U2 or UV (see Annex 2.H) symbols shown in Annex 2.F will be granted by ASME pursuant to the provisions of the following paragraphs. Stamps for applying the Code symbol shall be obtained from ASME.

2.G.2 Application for Authorization

2.G.2.1 Any organization desiring a Certificate of Authorization shall apply to ASME. The applications and related forms and information may be obtained from ASME Conformity Assessment Department, Three Park Avenue, New York, NY 10016-5990 or www.asme.org.

2.G.2.2 When an organization intends to build Code items in plants in more than one geographical area separate applications for each plant shall be submitted. Each application shall identify the accredited Authorized Inspection Agency providing Code inspection at each plant. A separate Certificate of Authorization will be issued for each plant.

2.G.2.3 Each applicant shall agree that each Certificate of Authorization and each Code symbol stamp are at all times the property of ASME, that they will be used in accordance with the requirements of this Division of the Code, and that they will be promptly returned to ASME upon request, or when the applicant discontinues the Code activities covered by this certificate, or when the Certificate of Authorization has expired and no new certificate has been issued. The holder of a Code symbol stamp shall not allow any other organization to use it.

2.G.3 Issuance of Authorization

2.G.3.1 Authorization to use the Code symbol stamps may be granted or withheld by ASME at its discretion. If authorization is granted and the proper administrative fee paid, a Certificate of Authorization evidencing permission to use any such symbol will be forwarded to the applicant. Each such certificate will identify the Code symbol to be used and the type of shop operations, field operations, or both for which authorization is granted.

2.G.3.2 Certificates are valid for not more than three years. Six months prior to the date of expiration of any such certificate, the applicant shall apply for a renewal of such authorization and the issuance of a new certificate.

2.G.3.3 ASME reserves the absolute right to cancel or refuse to renew such authorization.

2.G.3.4 ASME may at any time make such requirements concerning the issuance and use of Code symbol stamps as it deems appropriate, and all such requirements shall become binding upon the holders of valid Certificates of Authorization.

2.G.4 Inspection Agreement

2.G.4.1 As a condition of obtaining and maintaining a Certificate of Authorization to use the U2 Code symbol stamp, the Manufacturer shall have in force at all times an inspection contract or agreement with an ASME accredited Authorized Inspection Agency to provide inspection services. This inspection agreement is a written agreement between the Manufacturer and the Inspection Agency which specifies the terms and conditions under which the inspection services are to be furnished and which states the mutual responsibilities of the Manufacturer and the Authorized Inspectors. A certificate holder shall notify ASME whenever his agreement with an accredited Authorized Inspection Agency is canceled or changed to another accredited Authorized Inspection Agency.

2.G.4.2 Manufacturers of pressure relief valves are not required to have an inspection agreement with an accredited Authorized Inspection Agency.

2.G.5 Quality Control System

Any Manufacturer holding or applying for a Certificate of Authorization to use the U2 or UV stamp shall demonstrate a Quality Control System to establish that all Code requirements shall be met. The Quality Control System shall be in accordance with the requirements of Annex 2.E. A written description or checklist of the Quality Control System which identifies what documents and what procedures the Manufacturer will use to produce a Code item shall be available for review.

2.G.6 Evaluation for Authorization and Reauthorization

2.G.6.1 Before issuance or renewal of a Certificate of Authorization for use of the U2 stamp, the Manufacturer’s facilities and organization are subject to a joint review by a representative of his inspection agency and an individual certified as an ASME designee who is selected by the concerned legal jurisdiction. For those areas where there is no jurisdiction or where a jurisdiction does not choose to select an ASME designee to review a Manufacturer’s facility, an ASME designee selected by ASME shall perform that function. Where the jurisdiction is the Manufacturer’s Inspection Agency, the jurisdiction and the ASME designee shall make the joint review and joint report.

2.G.6.2 The purpose of the review by an ASME designee is to evaluate the applicant’s Quality Control System and its implementation. The ASME Designee performs reviews, surveys, audits, and examinations of organizations or persons holding or applying for accreditation or certification in accordance with the ASME code or standard. The applicant shall demonstrate sufficient administrative and fabrication functions of the system to show that he has the knowledge and ability to produce the Code items covered by his Quality Control System. Fabrication functions may be demonstrated using current work, a mock-up, or a combination of the two.

2.G.6.3 A written report to ASME shall be made jointly by the ASME designee and the accredited Authorized Inspection Agency employed by the Manufacturer to do the Manufacturer’s Code inspection. This report is then reviewed by ASME, which will either issue a Certificate of Authorization or notify the applicant of deficiencies revealed by the review. In the latter case, the applicant will be given an opportunity to explain or correct these deficiencies.

2.G.6.4 Before issuance or renewal of a Certificate of Authorization for use of the UV stamp, the valve Manufacturer’s facilities and organization are subject to a review by an ASME designee. A written description or checklist of the Quality Control System, which identifies the documents and procedures the Manufacturer will use to produce Code pressure relief valves, shall be available for review. The ASME designee shall make a written report to ASME, which will act on it as described in 2.G.6.3.

2.G.6.5 The Manufacturer may at any time make changes in the Quality Control System concerning the methods of achieving results, subject to acceptance by the Authorized Inspector. For Manufacturers and of UV stamped pressure relief valves, such acceptance shall be by the ASME designee.

2.G.7 Code Construction Before Receipt of Certificate of Authorization

A Manufacturer may start fabricating Code items before receipt of a Certificate of Authorization to use a Code symbol stamp under the following conditions:

a) The fabrication is done with the participation of the Authorized Inspector and is subject to his acceptance,

b) The activity is in conformance with the applicant's Quality Control System, and;

c) The item is stamped with the appropriate Code symbol and certified once the applicant receives his Certificate of Authorization from ASME.

ANNEX 2.H

GUIDE TO INFORMATION APPEARING ON THE CERTIFICATE OF AUTHORIZATION

(INFORMATIVE)

2.H.1 Introduction

2.H.1.1 The instructions in this Annex provide guidance in preparing a Certificate of Authorization.

2.H.1.2 The instructions for completing the Certificate of Authorization are identified by numbers corresponding to numbers on the sample forms in this Annex (see Table 2.H.1 and Figure 2.H.1).

2.H.2 Tables

Table 2.H.1 – Instructions For The Preparation Of A Certificate Of Authorization

Note No.	Instructions
1	The name of the Manufacturer or Assembler. The full street address, city, state or province, country, and zip code.
2	This entry describes the scope and limitations, if any, on use of the Code Symbol Stamps. Illustrated below are some examples of scope statements. <u>U2 Code Symbol Stamp</u> 1. Manufacture of pressure vessels at the above location only 2. Manufacture of pressure vessels at the above location only (This authorization does not cover welding or brazing) 3. Manufacture of pressure vessels at the above location and field sites controlled by the above location. 4. Manufacture of pressure vessels at the above location and field sites controlled by the above location (This authorization does not cover welding or brazing) 5. Manufacture of pressure vessels at field sites controlled by the above location 6. Manufacture of pressure vessels at field sites controlled by the above location (This authorization does not cover welding or brazing) 7. Manufacture of pressure vessel parts at the above location only. 8. Manufacture of pressure vessel parts at the above location and field sites controlled by the above location. 9. Manufacture of pressure vessel parts at field sites controlled by the above location. <u>UV Code Symbol Stamp</u> 1. Manufacture of pressure vessel pressure relief valves at the above location only 2. Manufacture of pressure vessel pressure relief valves at the above location only (This authorization does not cover welding or brazing) 3. Assembly of pressure vessel pressure relief valves at the above location (This authorization does not cover welding or brazing) 4. Manufacture of pressure vessel pressure relief valves and assembly of pressure vessel pressure relief valves at the above location only (The assembly of valves does not cover welding or brazing). 5. Manufacture of pressure vessel pressure relief valves and assembly of pressure vessel pressure relief valves at the above location only (This authorization does not cover welding or brazing). 6. Manufacture of pressure vessel pressure relief devices at the above location only.
3	The date authorization was granted by ASME to use the indicated Code symbol stamp.
4	The date authorization to use the Code symbol stamp will expire.
5	A unique certificate number assigned by ASME
6	Code symbol granted by ASME, i.e., U2 pressure vessels, UV pressure relief valves.
7	The signatures of the current chair.
8	The signatures of the current director.

2.H.3 Figures

The American Society of Mechanical Engineers

CERTIFICATE OF AUTHORIZATION

SYMBOL – 6

This certificate accredits the named company as authorized to use the Indicated symbol of the American Society of Mechanical Engineers (ASME), for the scope of activity shown below in accordance with the applicable rules of the ASME Boiler and Pressure Vessel Code. The use of the code symbol and the authority granted by this Certificate of Authorization are subject to the provisions of the agreement set forth in the application. Any construction stamped with this symbol shall have been built strictly in accordance with the provisions of the ASME Boiler and Pressure Vessel Code.

COMPANY – 1

SCOPE – 2

AUTHORIZED – 3

EXPIRES – 4

CERTIFICATE NUMBER – 5

7

____

CHAIRMAN OF THE BOILER AND PRESSURE VESSEL COMMITTEE

8

____

DIRECTOR, ASME ACCREDITATION

Figure 2.H.1 – Sample Certificate of Authorization

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PART 3

MATERIALS REQUIREMENTS

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3.1 General Requirements

The requirements for materials used in the construction of pressure vessel parts according to the rules of this Division are defined in this Part. General rules and supplemental requirements are defined for different material types and product forms. In cases of conflicts, the requirements stipulated in the paragraphs containing "Supplemental Requirements" shall govern.

3.2 Materials Permitted For Construction of Vessel Parts

3.2.1 Materials for Pressure Parts

3.2.1.1 Materials used for the construction of pressure parts shall conform to one of the specifications given in Section II, and shall be limited to those material specifications shown in the allowable design stress tables in Annex 3.A unless specifically allowed by other rules of this Division.

3.2.1.2 Materials outside the limits of size, thickness, or weight limits stipulated in the title or scope clause of the material specification given in Section II and permitted by paragraph 3.2.1.1 may be used if the material is in compliance with the other requirements of the specification and a size, thickness, or weight limitation is not given in the allowable design stress table (see Annex 3.A) or in Table 7.2. For specifications in which chemical composition or mechanical properties vary with size or thickness, materials outside the range shall be required to conform to the composition and mechanical properties shown for the nearest specified range.

3.2.1.3 Materials shall be proven of weldable quality. Satisfactory qualification of the welding procedure under Section IX is considered as proof.

3.2.1.4 Materials for which fatigue curves are provided (see paragraph 3.15) shall be used in construction of vessels or vessel parts subject to fatigue unless the fatigue analysis exemption criteria of paragraph 5.5.2 are satisfied.

3.2.1.5 Materials other than those allowed by this Division shall not be used unless data therein are submitted to and approved by the Boiler and Pressure Vessel Committee in accordance with Appendix 5 of Section II, Part D.

3.2.1.6 The rules in this Division do not provide detailed requirements for selection of an alloy suitable for the intended service or the amount of corrosion allowance to be provided. It is required that the user or his designated agent assure the materials used for the construction of vessels or vessel parts are suitable for the intended service conditions with respect to mechanical properties, resistance to corrosion, erosion, oxidation, and other damage mechanisms anticipated during service life. Informative and non-mandatory guidance regarding metallurgical phenomena that occur in material subject to certain process environments is provided in Appendix A of Section II, Part D.

3.2.1.7 The material specifications listed in Annex 3.A of this Division include a column of UNS (Unified Numbering System) numbers assigned to identify the various alloy compositions. These numbers are used in the rules of this Division whenever reference is made to materials of approximately the same chemical composition that are furnished under more than one approved specification or in more than one product form.

3.2.2 Materials for Attachments to Pressure Parts

3.2.2.1 Except as permitted in paragraph 3.2.2.2, materials for non-pressure parts which are welded to pressure parts shall meet all the requirements of paragraph 3.2.1 and all supplemental requirements stipulated in this Part (see Part 2, paragraph 2.2.2.1.g).

3.2.2.2 Except as limited in paragraph 3.5 for quenched and tempered steels, or by paragraph 6.7 for forged vessel construction where welding is not permitted, minor attachments may be of a non-ASME certified material and may be welded directly to the pressure part provided the criteria listed below are satisfied. In this context, minor attachments are parts of small size (i.e. not over 10 mm (3/8 in.) thick or 80 cm^3 (5 $in.^3$) volume) that support no load or insignificant loads (i.e. stress calculations are not required in the Manufacturer's judgment), such as name plates, insulation supports, and locating lugs.

a) The material is identified and is suitable for welding. Satisfactory qualification of welding procedure under Section IX is considered as proof.
b) The material is compatible insofar as welding is concerned with that to which the attachment is to be made.
c) The welds are postweld heat treated when required by Part 6, paragraph 6.4.2 of this Division.

3.2.3 Welding Materials

3.2.3.1 Welding materials used for the construction of pressure parts shall comply with the requirements of this Division, those of Section IX, and the applicable qualified welding procedure specification.

3.2.3.2 When the welding materials comply with one of the specifications in Section II, Part C, the marking or tagging of the material, containers, or packages as required by the applicable Section II specification may be adopted for identification in lieu of a Certified Test Report or a Certificate of Compliance. When the welding materials do not comply with one of the specifications of Section II, the marking or tagging shall be identifiable with the welding materials set forth in the welding procedure specification, and may be acceptable in lieu of a Certified Test Report or a Certificate of Compliance.

3.2.4 Dissimilar Materials

3.2.4.1 The user or his designated agent shall ensure that the coupling of dissimilar materials will not have a detrimental effect on the corrosion rate or service life of the vessel (see Appendix A of Section II, Part D).

3.2.4.2 The requirements for the base metals, heat affected zones (HAZ), and weld metal(s) of dissimilar metal weldments shall each be applied in accordance with the rules of this Division.

3.2.5 Product Specifications

3.2.5.1 The term plate as used in this Division also includes sheet and strip.

3.2.5.2 Rods and bars may be used in pressure vessel construction for pressure parts such as flange rings, stiffening rings, frames for reinforced openings, stays and staybolts, and similar parts. Except for flanges of all types, hollow cylindrically shaped parts [up to and including DN 100 (NPS 4)] may be machined from hot-rolled or forged bar provided that the axial length of the part is approximately parallel to the metal flow lines of the stock. Other parts, such as heads or caps [up to and including DN 100 (NPS 4)], not including flanges may be machined from hot-rolled or forged bar. Elbows, return bends, tees and header tees shall not be machined directly from bar stock.

3.2.5.3 When a material specification is not listed in this Division covering a particular wrought product of a grade, but there is an approved specification listed in this Division covering some other wrought product of that grade, the product for which there is no specification listed may be used provided:

a) The chemical and mechanical properties, heat treating requirements, and requirements for deoxidation, or grain size requirements conform to the approved specification listed in this Division. The stress values for that specification given in Annex 3.A shall be used.
b) The material specification is published Section II and covers that grade.
c) For the case of welded product forms without the addition of filler metal, the appropriate stress intensity values are multiplied by 0.85.
d) The product is not fabricated by fusion welding with the addition of filler metal unless it is fabricated in accordance with the rules of this Division as a pressure part.

e) The mill test reports reference the specifications used in producing the material and in addition make reference to this paragraph.

3.2.6 Certification

3.2.6.1 Certificate of Compliance and Material Test Report

a) The Manufacturer shall ensure all requirements of the material specification, and all special requirements of Part 3 of this Division, that are to be fulfilled by the materials manufacturer have been complied with. The Manufacturer shall accomplish this by obtaining Certificates of Compliance or Material Test Reports. These documents shall include results of all required tests and examinations, evidence of compliance with the material specifications and additional requirements as applicable. When the specification permits certain specific requirements to be completed later, those incomplete items shall be noted on the material documentation. When these specific requirements have been completed by someone other than the material manufacturer, this completion shall be documented and attached to the material documentation.
b) The manufacturer shall receive a copy of the test report as prepared by the originator of the data and maintain as part of his construction records.
c) All conflicts between the material specification and the supplemental requirements stipulated in this Part shall be noted, and compliance with the supplemental requirements shall be certified.

3.2.6.2 Certificate of Compliance and Material Test Reports by Other than Materials Manufacturer

a) Except as otherwise provided in paragraphs 3.2.5.3 and 3.2.7, if the requirements in a material specification listed in Annex 3.A have been completed by other than the materials manufacturer, then the vessel Manufacturer shall obtain supplementary material test reports and the Inspector shall examine these documents and determine that they represent the material and meet the requirements of the material specification.
b) The vessel Manufacturer shall certify compliance with all the supplemental requirements stipulated in this Part for any of the treatments or examinations specified herein. The certification shall include certified reports of results of all tests and examinations performed on the materials by the vessel Manufacturer.

3.2.7 Product Identification and Traceability

3.2.7.1 General Requirements

a) Material for pressure parts shall be organized so that when the vessel is completed, one complete set of the original identification markings required in the specifications for all materials of construction will be clearly visible. In case the original identification markings are unavoidably cut out or the material is divided into two or more parts, the vessel Manufacturer shall assure identification of each piece of material during fabrication and subsequent identification of the markings on the completed vessel by using the methods listed below.
    1) Accurate transfer of the original identification markings to a location where the markings will be visible on the completed vessel.
    2) Identification by coded marking, described in the Quality System Manual, acceptable to the Inspector and traceable to the original required marking.
b) An as-built sketch or tabulation of materials shall be made, identifying each piece of material with a certified test report or, where permitted by this Part, with a certificate of compliance and the coded marking which assure identification of each piece of material during fabrication and subsequent identification in the completed vessel.
c) When plate specification heat treatments are not performed by the material manufacturer, they shall be performed by, or under the control of, the vessel Manufacturer who shall then place the letter “T” following the letter “G” in the Mill plate marking (see SA-20) to indicate that the heat treatments required by the material specification have been performed. The fabricator shall also document in accordance with paragraph 3.2.6.2(b) that the specified heat treatments have been performed in accordance with the material manufacturer’s recommendation.

3.2.7.2 Method of Transferring Markings by the Manufacturer

a) Transfer of markings shall be made prior to cutting except that the Manufacturer may transfer markings immediately after cutting provided the control of these transfers is described in the Manufacturer's written Quality Control System. The Inspector need not witness the transfer of the marks but shall be satisfied that it has been done correctly.
b) The material may be marked by any method acceptable to the Inspector; however, all steel stamping shall be done with commercially available "low stress" dies.
c) Where the service conditions prohibit die-stamping for material identification, and when so specified by the user, the material manufacturer and the vessel Manufacturer shall mark the required data on the plates in a manner which will allow positive identification upon delivery. The markings must be recorded so that each plate will be positively identified in its position in the completed vessel to the satisfaction of the Inspector.

3.2.7.3 Transfer of Markings by Other Than the Manufacturer

a) When material is to be formed into shapes by anyone other than the Manufacturer of the completed pressure vessel and the original markings as required by the applicable material specification are unavoidably cut out, or the material is divided into two or more parts, the manufacturer of the shape shall either:
   1) Transfer the original identification markings to another location on the shape.
   2) Provide for identification by the use of a coded marking traceable to the original required marking, using a marking method agreed upon and described in the Quality Control System of the Manufacturer of the completed pressure vessel.
b) The mill certification of the mechanical and chemical properties requirements of the material formed into shapes, in conjunction with the above modified marking requirements, shall be considered sufficient to identify these shapes. Manufacturer's Partial Data Reports and parts stamping are not required unless there has been fabrication of the shapes that include welding, except as exempted by paragraph 3.2.8.2.

3.2.7.4 Marking of Plates

The material manufacturer's identification marking required by the material specification shall not be stamped on plate material less than 6 mm (1/4 in.) in thickness unless the following requirements are met.
a) The materials shall be limited to P-No. 1 Group Nos. 1 and 2.
b) The minimum nominal plate thickness shall be 5 mm (3/16 in.) or the minimum nominal pipe wall thickness shall be 4 mm (0.154 in.).
c) The MDMT shall be no colder than -29°C (-20°F).

3.2.8 Prefabricated or Preformed Pressure Parts

3.2.8.1 General Requirements

a) Prefabricated or preformed parts which are subject to the working pressure in the vessel and that are furnished by other than the shop of the Manufacturer responsible for the completed vessel shall conform to all applicable requirements of this Division as related to the completed vessel, including inspection in the shop of the parts manufacturer and by furnishing of Partial Data Reports, except for miscellaneous parts as permitted in paragraphs 3.2.8.2 and 3.2.8.3.
b) When the prefabricated or preformed parts are furnished with a nameplate and the nameplate interferes with further fabrication or service, and where stamping on the material is prohibited, the Manufacturer of the completed vessel, with the concurrence with the Authorized Inspector, may remove the nameplate. The removal of the nameplate shall be noted in the Remarks section of the vessel Manufacturer's Data Report. The nameplate shall be destroyed.
c) The rules of paragraphs 3.2.8.2 and 3.2.8.3 shall not be applied to quick opening closures (see paragraph 4.8).

3.2.8.2 Cast, Forged, Rolled, or Die Formed Standard Pressure Parts

a) Standard pressure parts such as pipe fittings, flanges, nozzles, welding necks, welding caps, manhole frames, and covers that are wholly formed by casting, forging, rolling, or die forming shall not require

inspection, material certification in accordance with paragraph 3.2.6 or Partial Data Reports.

b) Standard pressure parts that comply with the referenced ASME/ANSI standard (see paragraph 1.3) shall be made of materials permitted under this Division or of materials specifically listed in the ASME/ANSI product standard. These are pressure parts that comply with an ASME/ANSI product standard and are listed in Table 1.1. The ASME/ANSI product standard establishes the basis for the pressure-temperature marking, unless modified by this Division.

c) Standard pressure parts that comply with a manufacturer's standard shall be made of materials permitted by this Division. These are pressure parts that comply with a parts manufacturer's standard that defines the pressure-temperature rating marked on the part and described in the parts manufacturer's literature. The Manufacturer of the completed vessel shall ensure that the part complies with the requirements of this Division for the design conditions specified for the completed vessel. The material shall meet the requirements of paragraph 3.2.6.

d) Parts made to either an ASME/ANSI Standard or a manufacturer's standard shall be marked with the name or trademark of the parts manufacturer and such other markings as are required by the standard. Such marking shall be considered as the parts manufacturer's certification that the product complies with the material specifications and standards indicated, and is suitable for service at the pressure-temperature rating except as limited by the rules in paragraph 4.1.11.2 of this Division. The intent of this paragraph will have been met if, in lieu of the detailed marking on the part itself, the parts described herein have been marked in any permanent or temporary manner that will serve to identify the part with the parts manufacturer's detailed listing of the particular items and such listings are available for examination by the Inspector.

3.2.8.3 Cast, Forged, Rolled, or Die Formed Nonstandard Pressure Parts

a) Nonstandard pressure parts such as shell components, heads, removable doors, and pipe coils that are wholly formed by casting, forging, rolling, or die forming may be supplied basically as materials. All such parts shall be made of materials permitted by this Division and the manufacturer of that part shall furnish materials certification in accordance with paragraph 3.2.6.1. Such parts shall be marked with the name or trademark of the manufacturer and with such other as will serve to identify the particular parts with accompanying material certification.

b) The Manufacturer of the completed vessel shall ensure that the part complies with the requirements of this Division for the design conditions specified for the completed vessel.

3.2.8.4 Prefabricated or preformed pressure parts welded with filler metal shall be constructed in accordance with the rules of this Division.

3.2.9 Definition of Product Form Thickness

3.2.9.1 The requirements in this Division make reference to thickness. When the material specification does not specify thickness, the following definitions of nominal thickness apply.

a) Plate – the thickness is the dimension of the short transverse dimension.

b) Forgings – the thickness is the dimension defined as follows:

   1) Hollow Forgings – the nominal thickness is measured between the inside and the outside surfaces (radial thickness).

   2) Disk Forgings– the nominal thickness is the axial length (axial length ≤ outside the diameter).

   3) Flat Ring Forgings – for axial length less than or equal to 50 mm (2 in.), the axial length is the nominal thickness; for axial length greater than 50 mm (2 in.), the radial thickness is the nominal thickness (axial length less than the radial thickness).

   4) Rectangular Solid Forgings – the least rectangular dimension is the nominal thickness.

   5) Round, Hexagonal and Octagonal Solid Forgings - the nominal thickness is the diamter or distance across the flats (axial length > diameter or distance across the flats).

c) Castings – for castings of the general shapes described for forgings, the same definitions apply. For other castings, the maximum thickness between two cast coincidental surfaces is the nominal thickness.

3.2.9.2 The definition of nominal thickness for postweld heat treat requirements is covered in paragraph 6.4.2.7.

3.2.10 Product Form Tolerances

3.2.10.1 Plate

Plate material shall be ordered not thinner than the design thickness. Vessels made of plate furnished with an undertolerance of not more than the smaller value of 0.3 mm (0.01 in.) or 6% of the ordered thickness may be used at the full design pressure for the thickness ordered if the material specification permits such an undertolerance. If the specification to which the plate is ordered allows a greater undertolerance, the ordered thickness of the material shall be sufficiently greater than the design thickness so that the thickness of the material furnished is not more than the smaller of 0.3 mm (0.01 in.) or 6% under the design thickness.

3.2.10.2 Pipe and Tube

If pipe or tube is ordered by its nominal wall thickness, the manufacturing undertolerance on wall thickness shall be taken into account. After the minimum required wall thickness is determined, it shall be increased by an amount sufficient to provide the manufacturing undertolerance allowed in the pipe or tube specification.

3.2.11 Purchase Requirements

3.2.11.1 A summary of the pertinent requirements in paragraphs 3.2 through 3.8 is provided in Annex 3.B.

3.2.11.2 Special chemical compositions, heat treatment procedures, fabrication requirements, and supplementary tests may be required to assure that the vessel will be in the most favorable condition for the intended service.

3.3 Supplemental Requirements for Ferrous Materials

3.3.1 General

All forms of ferrous products listed in Table 3.A.1 shall meet the supplemental requirements of paragraph 3.3. The high strength quenched and tempered steels listed in Table 3.A.2, shall meet the supplemental requirements of paragraph 3.4.

3.3.2 Chemistry Requirements

Carbon and low alloy steel having carbon content of more than 0.35% by heat analysis shall not be used in welded construction or be shaped by oxygen cutting (except as provided elsewhere in this Division).

3.3.3 Ultrasonic Examination of Plates

3.3.3.1 Except as permitted in paragraph 3.3.3.2, all plate 50 mm (2 in.) and over in nominal thickness shall be ultrasonically examined in accordance with the requirements of SA-578. The acceptance standard shall be Level B of SA-578.

3.3.3.2 When the design rules permit credit for thickness of cladding on plate conforming to SA-263, SA-264, and SA-265, ultrasonic examination shall be made of the base plate and the bond between the cladding and the base plate in accordance with SA-578 with acceptance criteria of S-6 of Supplementary Requirements regardless of the thickness of plate.

3.3.4 Ultrasonic Examination of Forgings

3.3.4.1 All forgings 50 mm (2 in.) and over in nominal thickness shall be examined ultrasonically in accordance with SA-388.

a) Rings, flanges, and other hollow forgings shall be examined using the angle beam technique. For other forgings, the straight beam technique shall be used.
b) Reference specimens shall have the same nominal thickness, composition, and P-number grouping as the forgings to be examined in order to have substantially the same structure.

3.3.4.2 Forgings are unacceptable if:

a) The straight beam examination results show one or more discontinuities which produce indications

accompanied by a complete loss of back reflection not associated with or attributable to the geometric configuration.

b) Angle beam examination results show one or more discontinuities which produce indications exceeding in amplitude the indication from the calibration notch.

3.3.4.3 In the case of straight beam examination, the following conditions shall be reported to the purchaser for his consideration and approval prior to shipment of the forging:

a) Forgings containing one or more indications with amplitudes exceeding adjacent back reflections.
b) Forgings containing one or more discontinuities which produce traveling indications accompanied by reduced back reflections. A traveling indication is defined as an indication that displays sweep movement of the oscilloscope screen at constant amplitudes as the transducer is moved.

3.3.4.4 In the case of angle beam examination, the following conditions shall be reported to the purchaser for his consideration and approval prior to shipment of the forging:

a) Indications having an amplitude exceeding 50% of the calibration block amplitude.
b) Clusters of indications located in a small area of the forging with amplitudes less than 50% of the calibration notch amplitude. A cluster of indications is defined as three or more indications exceeding 10% of the standard calibration notch amplitude and located in any volume approximately a 50 mm (2 in.) or smaller cube.

3.3.4.5 Additional nondestructive examination procedures or trepanning may be employed to resolve questions of interpretation of ultrasonic indications.

3.3.5 Magnetic Particle and Liquid Penetrant Examination of Forgings

3.3.5.1 Following final machining by the manufacturer all accessible surfaces of thick or complex forgings, such as contour nozzles, thick tubesheets, flanges, and other complex forgings that are contour shaped or machined to essentially the finished product configuration prior to heat treatment, shall be examined by the magnetic particle method in accordance with Test Method A 275/A 275M or by the liquid penetrant method in accordance with Practice E 165. The evaluation of indications detected by the magnetic particle method or by the liquid penetrant method and the acceptance standards shall be in accordance with Part 7 of this Division.

3.3.5.2 Unacceptable imperfections shall be removed and the areas shall be reexamined to ensure complete removal of the unacceptable imperfection. Unless prohibited by the material specification, the forgings may be repair welded with the approval of the vessel Manufacturer. Repairs shall be made utilizing welding procedures that have been qualified in accordance with Section IX. The repaired forging shall meet all requirements of this Division.

3.3.6 Integral and Weld Metal Overlay Clad Base Metal

3.3.6.1 Applied Linings

Material used for applied corrosion resistant lining may be any metallic material of weldable quality, provided all applicable requirements of this Division are satisfied.

3.3.6.2 Design Calculations Based on Total Thickness

Base material with corrosion resistant integral or weld metal overlay cladding used in construction in which the design calculations are based on total thickness including cladding (paragraph 4.1.9) shall consist of base plate listed in one of the material tables in Part 3 and shall conform to one of the following specifications or utilize weld metal overlay cladding meeting the requirements of this Division.

a) SA-263 Specification for Corrosion-Resisting Chromium-Steel Clad Plate, Sheet and Strip;
b) SA-264 Specification for Corrosion-Resisting Chromium-Nickel Steel Clad Plate, Sheet and Strip; or
c) SA-265 Specification for Nickel and Nickel-Base Alloy Clad Steel Plate.

3.3.6.3 Design Calculations Based on Base-Plate Thickness

Clad plate used in constructions in which the design calculations are based on the base-plate thickness,

exclusive of the thickness of the cladding material, may consist of any base-plate material satisfying the requirements of Part 3 and any metallic integral or weld metal overlay cladding material of weldable quality that meets the requirements of paragraph 6.5 of this Division.

3.3.6.4 Shear Strength of Bond of Integrally Clad Plates

Integrally clad plates in which any part of the cladding is included in the design calculations, as permitted in paragraph 4.1.9, shall show a minimum shear strength of 140 MPa (20 ksi) when tested in the manner described in the plate specification. One shear test shall be made on each such clad plate and the results shall be reported on the certified test report. A shear or bond strength test is not required for weld metal overlay cladding.

3.3.6.5 Removal of Cladding for Mill Tension Tests

When any part of the cladding thickness is specified an allowance for corrosion, such added thickness shall be removed before mill tension tests.

3.4 Supplemental Requirements for Cr–Mo Steels

3.4.1 General

3.4.1.1 The rules in paragraph 3.4 include supplemental requirements for fabrication and testing for Cr-Mo steels. The materials and appropriate specifications covered by this paragraph are listed in Table 3.1.

3.4.1.2 Certification that the requirements of paragraph 3.4 have been satisfied shall be shown on the Manufacturer's Data Report Form.

3.4.2 Postweld Heat Treatment

The final postweld heat treatment shall be in accordance with the requirements of paragraph 6.4.2 of this Division.

3.4.3 Test Specimen Heat Treatment

3.4.3.1 Two sets of tension specimens and one set of Charpy impact specimens shall be tested. One set each of the tension specimens shall be exposed to heat treatment condition A. The second set of tension specimens and the set of Charpy specimens shall be exposed to heat treatment Condition B.

a) Condition A – Temperature shall be no lower than the actual maximum vessel-portion temperature, less 14°C (25°F). Time at temperature shall be no less than 80% of the actual holding time of the vessel portion exposed to the maximum vessel-portion temperature.
b) Condition B – Temperature shall be no higher than the actual minimum vessel-portion temperature, plus 14°C (25°F). Time at temperature shall be no more than 120% of the actual hold time of the vessel portion exposed to the minimum vessel-portion temperature.

3.4.3.2 The suggested procedure for establishing the test specimen heat treatment parameters are shown below.

a) Establish maximum and minimum temperatures and hold times for the vessel/component heat treatment based on experience/equipment;
b) Determine Conditions A and B for the test specimen heat treatments;
c) Vessel heat treatment temperature and hold time limitations, and test specimen Conditions A and B, are shown in Figure 3.1.

3.4.4 Weld Procedure Qualifications and Weld Consumables Testing

3.4.4.1 Welding procedure qualifications using production weld consumables shall be made for material welded to itself or to other materials. The qualifications shall conform to the requirements of Section IX, and the maximum tensile strength at room temperature shall be 760 MPa (110 ksi) (for heat treatment Conditions A and B). Welding shall be limited to submerged-arc (SAW) and shielded metal-arc (SMAW) processes for 3Cr-1Mo-1/4V-Ti-B material only.

3.4.4.2 Weld metal from each heat or lot of electrodes and filler wire-flux combination shall be tested. The minimum and maximum tensile properties shall be met in postweld heat treated (PWHT) Conditions A and B. The minimum Charpy V-notch impact properties shall be met in PWHT Condition B. Testing shall be in general conformance with SFA-5.5 for covered electrodes and SFA-5.23 for filler wire-flux combinations.

3.4.4.3 Duplicate testing in PWHT Condition A and PWHT Condition B (see paragraph 3.4.3) is required. The minimum tensile strength and Charpy impact properties for the base metal shall be met. Charpy impact testing is only required for Condition B.

3.4.4.4 For 2¼Cr-1Mo-¼V material, the weld metal shall meet the compositional requirements listed in Table 3.2. For all other materials, the minimum carbon content of the weld metal shall be 0.05%.

3.4.4.5 In addition for 2¼Cr-1Mo and 2¼Cr-1Mo-¼V material, Category A welds intended for design temperatures above 440°C (825°F), each heat of filler wire and flux combination used in production shall also be qualified by a weld metal stress-rupture test on specimens machined parallel (all weld metal specimens) and transverse to the weld axis (one specimen each) in accordance with the following:

a) The specimen diameter within the gage length shall be 13 mm (1/2 in.) or greater. The specimen centerline shall be located at the 0.25-t thickness location (or closer to the center) for material 19 mm (3/4 in.) and greater in thickness.
b) The gage length for the transverse specimen shall include the weld and at least 19 mm (3/4 in.) of base metal adjacent to the fusion line.
c) The test material shall be postweld heat treated to Condition A.
d) For 2¼Cr-1Mo material, the condition of the stress-rupture test shall be 210 MPa (30 ksi) at 510°C (950°F). The time of failure shall exceed 650 hr.
e) For 2¼Cr-1Mo-¼V material, the condition of the stress-rupture test shall be 210 MPa (30 ksi) at 540°C (1000°F). The time of failure shall exceed 900 hr.

3.4.5 Toughness Requirements

The minimum toughness requirements for base metal, weld metal, and heat affected zone, after exposure to the simulated postweld heat treatment Condition B, are shown in Table 3.3. If the material specification or other parts of this Division have more demanding toughness requirements, they shall be met.

3.4.6 Ultrasonic Examination

For SAW weld in 2¼Cr-1Mo-V vessels, ultrasonic examination in accordance with 7.5.5 are required.
NOTE: for SAW welds in 2¼Cr-1Mo-V vessels, the potential exists for fabrication related cracking. Careful selection of examination techniques, scans, calibrations, and acceptance criteria are necessary to provide the sensitivity required to detect this cracking.

3.5 Supplemental Requirements for Q&T Steels with Enhanced Tensile Properties

3.5.1 General

3.5.1.1 The supplemental requirements in paragraph 3.5 apply to ferritic steels with tensile properties enhanced by quenching and tempering and shall be used in conjunction with the other requirements of this Division. The material specifications for these steels are shown in Table 3.A.2.

3.5.1.2 The requirements of this paragraph are not intended to apply to steels listed in Table 3.A.1 that are furnished in such thicknesses that heat treatment, involving the use of accelerated cooling, including liquid quenching, is used to obtain structures comparable to those attained by normalizing thinner sections.

3.5.2 Parts for Which Q&T Steels May be Used

High strength quenched and tempered steels shown in Table 3.A.2, may be used for the entire vessel or for individual components of vessels that are joined to other grades of quenched and tempered steels, or to other steels conforming to specifications listed in Tables 3.A.1, 3.A.3, and 3.A.6, subject to the requirements and limitations of this Division.

3.5.3 Structural Attachments

3.5.3.1 Except as permitted in paragraph 3.5.3.2 below, all permanent structural attachments and stiffening rings that are welded directly to pressure parts shall be made of material whose specified minimum yield strength is within ±20% of that of the material to which they are attached.

3.5.3.2 All permanent structural attachments welded directly to a shell or head constructed of a material conforming to SA-333, Grade 8, SA-334, Grade 8, SA-353, SA-522, SA-553, and SA-645 Grade A shall be made from a material covered by these same specifications, or nickel alloys UNS N06625 or N10276, or from wrought non-hardenable austenitic stainless steels. If an austenitic stainless steel is used, consideration should be given to the additional weld stresses resulting from the difference in thermal expansion between the attachment and the shell.

3.6 Supplemental Requirements for Nonferrous Materials

3.6.1 General

Nonferrous materials covered by paragraph 3.6 shall conform to one of the specifications listed in Tables 3.A.4, 3.A.5, 3.A.6, and 3.A.7, and shall be used in conjunction with the other requirements of this Division.

3.6.2 Ultrasonic Examination of Plates

All plates 50 mm (2 in.) and over in nominal thickness shall be ultrasonically examined in accordance with the applicable requirements of the ASTM standards and ASME specifications listed below:

a) SE-114 Ultrasonic Testing by Reflection Method Using Pulsed Longitudinal Waves Induced by Direct Contact;
b) E 214 Immersed Ultrasonic Testing by the Reflection Method Using Pulsed Longitudinal Waves;
c) E 127 Fabricating and Checking Aluminum Alloy Ultrasonic Standard Reference Blocks;
d) SB-548 Ultrasonic Testing of Aluminum Plate.

3.6.3 Ultrasonic Examination of Forgings

3.6.3.1 Insofar as practicable, all solid rectangular forgings shall be examined by the straight beam technique from two directions at approximately right angles. Hollow forgings including flanges and rings 50 mm (2 in.) and over in nominal thickness shall be examined using the angle beam technique by either the contact method or the immersion method. Reference specimens and acceptance criteria shall be examined from one face or surface normal to the axis in the circumferential direction unless the wall thickness or geometric configuration makes angle beam examination impracticable. Disk forgings shall be examined from one flat side and from the circumferential surface.

3.6.3.2 The entire volume of metal shall be ultrasonically examined at some state of manufacture. For heat treated material, examination after final heat treatment is preferred, but if the contour of the forging precludes complete examination at this stage, the maximum possible volume of the forging shall be reexamined after the final heat treatment.

3.6.3.3 The method used in the examination of forgings shall conform to the following requirements.

a) In straight beam examination, the transducers shall be 19 mm to 29 mm (3/4 in. to 1-1/8 in.) in diameter or 25 mm (1 in.) square. The nominal frequency shall be appropriate for the material being examined. The instrument shall be set so that the first back reflection is 75 ± 5% of the screen height when the transducer is placed on the indication-free area of the forging.
b) In angle beam examination by the contact method, a 25 mm x 25 mm (1 in. x 1 in.) or 25 mm x 38 mm (1 in. x 1-1/2 in.), 45 degree transducer shall be used at an appropriate frequency.
c) In angle beam examination by the immersion method, a 19 mm (3/4 in.) diameter transducer oriented at an approximate angle of inclination shall be used at an appropriate frequency.
d) Angle beam examination shall be calibrated with a notch of a depth equal to the smaller of 10 mm (3/8 in.) or 3% of the nominal section thickness, a length of approximately 25 mm (1 in.) and width not greater than two times the depth.

3.6.3.4 The material shall be unacceptable (unless repaired in accordance with the rules of this Division) if straight beam examination shows one or more discontinuities which produce indications accompanied by a complete loss of back reflection not associated with or attributable to the geometric configuration, or if angle beam examination results show one or more discontinuities which produce indications exceeding that of the calibration notch.

3.6.4 Liquid Penetrant Examination of Forgings

3.6.4.1 Following final machining by the manufacturer all accessible surfaces of thick and complex forgings, such as contour nozzles, thick tubesheets, flanges, and other complex forgings that are contour shaped or machined to essentially the finished product configuration prior to heat treatment, shall be examined by the liquid penetrant method in accordance with Practice E 165.

3.6.4.2 The evaluation of indications detected by the liquid penetrant method and the acceptance standards shall be in accordance with Part 7 of this Division.

3.6.4.3 Unacceptable imperfections shall be removed and the areas shall be reexamined to ensure complete removal of the unacceptable imperfection. Unless prohibited by the material specification, the forgings may be repair welded with the approval of the vessel Manufacturer. Repairs shall be made utilizing welding procedures that have been qualified in accordance with Section IX. The repaired forging shall meet all requirements of this Division.

3.6.5 Clad Plate and Products

Clad plate or products used in construction for which the design calculations are based on total thickness, including cladding, shall consist of base plate listed in one of the material tables in this Division and shall conform to one of the following specifications:

a) SB-209 Specification for Aluminum Alloy Sheet and Plate,

b) SB-211 Specification for Aluminum Alloy Extruded Bars, Rods, Shapes, and Tubes.

3.7 Supplemental Requirements for Bolting

3.7.1 General

The supplemental requirements in paragraph 3.7 are required for all bolts, studs, and nuts supplied with vessels constructed to this Division.

3.7.2 Examination of Bolts, Studs, and Nuts

Bolts, studs, and nuts covered by the material specifications listed in Annex 3.A shall be subjected to the following examinations:

a) All areas of threads, shanks, and heads of final machined parts shall be visually examined. Discontinuities, such as laps, seams, cracks are unacceptable.

b) All bolts, studs, and nuts over 25 mm (1 in.) nominal bolt size shall be examined by the magnetic particle method or by the liquid penetrant method in accordance with Part 7 of this Division. This examination shall be performed on the finished component after threading or on the material stock at approximately the finished diameter before threading and after heading (if involved). Linear non-axial indications are unacceptable. Linear indications greater than 25 mm (1 in.) in length are unacceptable.

c) All bolts, studs, and nuts greater than 50 mm (2 in.) nominal thickness shall be ultrasonically examined over the entire surface prior to threading in accordance with the following requirements:

   1) Examination shall be carried out by the straight beam, radial scan method.

   2) Examination shall be performed at a nominal frequency of 2.25 MHz with the search unit not to exceed 645 mm^2 (1 in^2) in area.

   3) Calibration sensitivity shall be established by adjustment of the instrument so that the first back screen reflection is 75 – 90% of full screen height.

4) Any discontinuity which causes an indication in excess of 20% of the height of the first back reflection or any discontinuity which prevents the production of the first back reflection of 50% of the calibration amplitude is not acceptable.

d) All bolts, studs, and nuts greater than 100 mm (4 in.) nominal diameter shall be ultrasonically examined over an entire end surface before or after threading in accordance with the following requirements:

1) Examination shall be carried out by the straight beam, longitudinal scan method.

2) Examination shall be performed at a nominal frequency of 2.25 MHz with the search unit not to exceed 320 mm^2 (0.5 $in.^2$) in area.

3) Calibration shall be established on a test bar of the same nominal composition and diameter as the production part and a minimum of one half of the length. A 10 mm (3/8 in.) diameter x 76 mm (3 in.) deep flat bottom hole shall be drilled in one end of the bar and plugged to full depth. A distance amplitude correction curve shall be established by scanning from both ends of the test bar.

4) Any discontinuity which causes an indication in excess of that produced by the calibration hole in the reference specimen as corrected by the distance amplitude correction curve is not acceptable.

3.7.3 Threading and Machining of Studs

3.7.3.1 Studs shall be threaded the full length, or shall be machined down to the root diameter of the thread in the unthreaded portion provided that the threaded portions are at least 1.5 diameters in length.

3.7.3.2 Studs greater than 8 diameters in length may have an unthreaded portion which has the nominal diameter of the thread, provided the following requirements are met:

a) The threaded portion shall be at least 1.5 diameters in length.
b) The stud shall be machined down to the root diameter of the thread for a minimum distance of 0.5 diameters adjacent to the threaded portion.
c) Suitable transition shall be provided between the root diameter and the unthreaded portion.
d) Particular consideration shall be given to any dynamic loadings.

3.7.4 Use of Washers

When washers are used in conjunction with torquing methods (e.g. the use of manual or hydraulic torque wrenches) for the purpose of bolt tightening, they shall be designed to provide a smooth and low-friction contact surface for the nuts, which are important considerations when torquing methods are used for bolt tightening. NOTE: Flat washers typically should be 6 mm (1/4 in.) thick and made of through-hardened, wrought low alloy steel. See ASME PCC-1 for more information.

3.7.5 Ferrous Bolting

3.7.5.1 Material for Ferrous Bolting

a) Approved specifications for ferrous bolting are given in Annex 3.A, Tables 3.A.8, 3.A.9, 3.A.10 and 3.A.11.
b) High alloy steel studs, bolts and nuts may be used with carbon and low alloy steel components, provided they are suitable for the application (see Section **II**, Part D, paragraph A-300, Metallurgical Phenomena).
c) Nonferrous nuts and washers may be used with ferrous bolts and studs, provided they are suitable for the application. Consideration shall be given to the differences in thermal expansion and possible corrosion resulting from combination of dissimilar materials.

3.7.5.2 Material for Ferrous Nuts and Washers

a) Material for nuts and washers shall conform to SA-194, SA-563, or to the requirements for nuts in the specification for the bolting material with which they are to be used.
b) Materials for ferrous nuts and washers shall be selected as follows:

1) Carbon or low alloy steel nuts and carbon or low alloy steel washers of approximately the same hardness as the nuts may be used for metal temperatures not exceeding 480°C (900°F).

2) Alloy steel nuts shall be used for metal temperatures exceeding 480°C (900°F). Washers, if used, shall be of alloy steel equivalent to the nut material.

3.7.5.3 Requirements for Ferrous Nuts

a) Nuts shall be semi-finished, chamfered, and trimmed. Nuts shall be threaded to Class 2B or finer tolerances according to ASME B1.1.
b) For use with flanges conforming to ASME/ANSI B16.5, nuts shall conform to at least to the dimensions given in ASME/ANSI B18.2.2 for Heavy Series Nuts.
c) For use with connections designed in accordance with rules in paragraph 4.16, nuts may be of the American National Standard Heavy Series or they may be of other dimensions provided their strength is equal to that of the bolting, giving due consideration to the bolt hole clearance, bearing area, thread form and class of it, thread shear, and radial thrust from threads.
d) Nuts shall engage the threads for the full depth of the nut or, in the case of cap nuts, to a depth equivalent to the depth of a standard nut.
e) Nuts of special design may be used provided their strength is equal to that of the bolting.

3.7.6 Nonferrous Bolting

3.7.6.1 Material for Nonferrous Bolting

Approved specifications for Nonferrous bolting are given i*n* Annex 3.A, Tables 3.A.9 and 3.A.10, and 3.A.11.

3.7.6.2 Condition of Material Selected and Allowable Stress Value

a) When nonferrous bolts are machined from heat treated, hot rolled, or cold worked material and are not subsequently hot worked or annealed, the allowable design stress values in Table 3 of Section II, Part D to be used in design shall be based on the condition of material selected.
b) When nonferrous bolts are fabricated by hot heading, the allowable design stress values for annealed materials in Table 3 of Section II, Part D shall apply unless the manufacturer can furnish adequate control data to show that the tensile properties of hot rolled or heat treated bars or hot finished or heat treated forgings are being met, in which case the allowable stress values for the material in the hot finished condition may be used.
c) When nonferrous bolts are fabricated by cold heading, the allowable design stress values for annealed materials in Table 3 of Section II, Part D shall apply unless the manufacturer can furnish adequate control data to show that higher design stresses, as agreed upon may be used. In no case shall such stresses exceed the allowable stress values given in Table 3 of Section II, Part D for cold worked bar stock.

3.7.6.3 Materials for Nonferrous Nuts and Washers

a) Materials for ferrous nuts used with nonferrous bolting shall conform to paragraph 3.7.5.3.
b) Nonferrous nuts and washers may be made of any suitable material listed in Tables 3.A.5, 3.A.6, and 3.A.7.

3.7.6.4 Requirements for Nonferrous Nuts

Nonferrous nuts shall meet the requirements in paragraph 3.7.5.3.

3.7.7 Materials for Ferrous and Nonferrous Nuts of Special Design

Nuts of special design, such as wing nuts, may be made of any suitable wrought material permitted by this Division, and shall be either: hot or cold forged; or machined from hot-forged, hot-rolled, or cold-drawn bars.

3.8 Supplemental Requirements for Castings

3.8.1 General

3.8.1.1 Each casting shall be marked with the name, trademark, or other traceable identification of the manufacturer and the casting identification, including material designation. The casting manufacturer shall furnish certification that each casting conforms to all the applicable requirements in the casting specification and the requirements of this Division. The certification of castings shall also indicate the nature, location, and extent of any repairs.

3.8.1.2 All castings to be welded shall be of weldable grade.

3.8.2 Requirements for Ferrous Castings

3.8.2.1 Centrifugal Steel Castings

In addition to the minimum requirements of the material specification, all surfaces of centrifugal castings shall be machined after heat treatment to a finish not coarser than 6.35 μmm (250 μin) arithmetic average deviation.

3.8.2.2 Nondestructive Examination of Ferrous Castings

a) General – Castings shall be examined by radiographic, ultrasonic, magnetic particle and liquid penetrant methods examination as provided herein and shall meet the requirements of paragraphs 3.8.2.2.a through 3.8.2.2.d, inclusive. Radiographic examination, and when required ultrasonic examination, of castings shall be made after at least one austenitizing heat treatment, except austenitic castings not requiring heat treatment may have radiographic and ultrasonic examination performed at any stage of manufacture. Magnetic particle or liquid penetrant examinations shall be made after final heat treatment and after final machining of machined areas.

b) Radiographic Examination – All parts of ferrous castings regardless of thickness shall be fully radiographed in accordance with the procedures of Article 2 of Section V. The radiographs shall be compared to the appropriate Radiographic Standard listed below, and the maximum acceptable severity levels for imperfection shall be as follows:

   1) For castings having radiographed thickness of less than 50 mm (2 in.), E 446, Standard Reference Radiographs For Steel Castings Up To 50 mm (2 in.) in thickness, and with maximum severity levels as shown in Table 3.9.

   2) For castings having radiographed thickness from 50 mm to 305 mm (2 in. to 12 in.), E 186, Standard Reference Radiographs for Heavy-Walled 50 mm to 115 mm (2 in. to 4.5 in.) Steel Castings or E 280, Standard Reference Radiographs for Heavy-Walled 115 mm to 305 mm (4.5 in. to 12 in.) Steel Castings, as appropriate, and with maximum severity levels as shown in Table 3.10.

c) Ultrasonic Examination – All parts of ferrous castings over 305 mm (12 in.) thick shall be examined by ultrasonic methods in accordance with the procedures of Article 5 of Section V. Castings with imperfections shown by discontinuities whose reflections exceed the height equal to 20% of the normal back reflection, or which reduce the height of the back reflections by more than 30% during movement of the transducer 50 mm (2 in.) in any direction are unacceptable unless other methods of nondestructive testing, such as radiographic examination, demonstrate to the satisfaction of the vessel Manufacturer and the Inspector that the indications are acceptable or unless such imperfections are removed and the casting is repaired.

d) Magnetic Particle Examination – Castings of magnetic material shall be examined on all surfaces by a magnetic particle method in accordance with Part 7 of this Division. Castings with imperfections shown by Type I indications or by indications exceeding Degree I of Types II, III, IV, and V of ASTM E 125, Reference Photographs for Magnetic Particle Indications on Ferrous Castings, are unacceptable unless the imperfections are removed and casting is repaired.

e) Liquid Penetrant Examination – Castings of nonmagnetic material shall be examined on all surfaces by a liquid penetrant method in accordance with Part 7 of this Division. Castings with cracks and linear imperfections exceeding the following limits are unacceptable:

   1) Linear indications resulting in more than six indications in any 40 mm x 150 mm (1-1/2 in x 6 in.) rectangle or 90 mm (3.5 in.) diameter circle with these taken in the most unfavorable location relative to the indications being evaluated.

   2) Linear imperfections resulting in indications more than 6 mm (1/4 in.) in length for thicknesses up to 19 mm (3/4 in.), one third of the thickness in length for thicknesses from 19 mm (3/4 in.) to 57 mm (2.25 in.), and 19 mm (3/4 in.) in length for thicknesses over 57 mm (2.25 in.). Aligned acceptable imperfections separated from one another by a distance equal to the length of the longer imperfection are acceptable.

   3) All nonlinear imperfections which are indicated to have any dimension which exceeds 2.5 mm (0.0938 in.).

3.8.2.3 Repairing of Ferrous Castings

a) Castings with unacceptable imperfections may be repaired. Whenever an imperfection is removed and subsequent repair by welding is not required, the affected area shall be blended into the surrounding surface so as to avoid sharp notches, crevices, or corners.

b) Repairing of Ferrous Castings by Welding – Castings having imperfections in excess of the maximum sizes permitted in paragraph 3.8.2.2 may be repaired by welding if the imperfections are removed and providing prior approval is obtained from the vessel Manufacturer. To ensure complete removal of such imperfections prior to making repairs the base metal shall be reexamined by either magnetic particle or liquid penetrant examination, if it is magnetic, or by liquid penetrant examination, if it is nonmagnetic.

   1) Requirements for Examining Repairs in Castings – All weld repairs of depth exceeding 10 mm (3/8 in.) or 20% of the section thickness, whichever is the lesser, shall be examined by radiography and by magnetic particle examination or liquid penetrant examination, if the material is magnetic, or by liquid penetrant examination, if it is nonmagnetic, in accordance with paragraph 3.8.2.2. Where the depth of the repairs is less than 20% of the section thickness or 25 mm (1 in.), whichever is the lesser, and where the repaired section cannot be radiographed effectively, the first layer of each 6 mm (1/4 in.) thickness of deposited weld metal and the finished weld surface shall be examined, as indicated previously by magnetic particle or liquid penetrant examination. The finished surface examination shall be made after any heat treating operations that are applied to the casting. Weld repairs resulting from ultrasonic examination shall be examined by ultrasonic methods.

   2) Postweld Heat Treatment of Repaired Castings – When repair welding is done after heat treatment of the casting, the casting shall be postweld heat treated after repair welding of the casting.

   3) Required Welding Procedure and Welder Qualifications – All welding shall be performed with a welding procedure qualified in accordance with Section IX. The procedure qualification tests shall be performed on specimens of cast material of the same specification and subject to the same heat treatment before and after welding as will be applied to the work. All welders and operators performing this welding shall also be qualified in accordance with Section IX.

   4) Certification of Weld Repairs – The location and extent of the weld repairs together with the repair procedure and examination results shall be recorded and transmitted as part of the certification.

3.8.3 Requirements for Nonferrous Castings

3.8.3.1 Examination of Nonferrous Castings

All nonferrous castings shall be examined in accordance with the following:

a) Each casting shall be subjected to 100% visual examination and to liquid penetrant examination on all surfaces in accordance with paragraph 3.8.2.2.e. These examinations shall be performed following the final heat treatment applied to the casting.

b) All parts of castings shall be subjected to complete radiographic examination and the radiographs shall be compared with the radiographic standards of ASTM E 272, Reference Radiographs for Inspection of High Strength Copper Base and Nickel-Copper Castings. Acceptable castings shall meet Class 1 standards, if the wall thickness is less than 25 mm (1 in.) or Class 2 standards if the wall thickness is greater than or equal to 25 mm (1 in.) as defined in the ASTM Specification.

c) All parts of castings with a thickness greater than 305 mm (12 in.) shall be ultrasonically examined in accordance with the procedures given in SE-114. Any imperfections whose reflections do not exceed a height equal to 20% of the normal back reflection or do not reduce the height of the back reflection by more than 30% during movement of the transducer 50 mm (2 in.), in any direction, shall be considered acceptable. The above limits are established for the use of transducers having approximately 645 mm^2 (1 in^2) of area.

3.8.3.2 Repairing of Nonferrous Castings by Welding

Upon approval by the vessel Manufacturer, castings subject to rejection because of these examinations may be repaired in accordance with the following requirements.

a) Castings having imperfections in excess of the maximum sizes permitted in paragraph 3.8.3.1 may be repaired by welding, if the imperfections are removed and provided prior approval is obtained from the vessel Manufacturer. To assure complete removal of such imperfections, prior to making repairs, the

base metal shall be reexamined by liquid penetrant examination.

b) All weld repairs of depth exceeding 10 mm (3/8 in.), or 20% of the section thickness, whichever is the lesser, shall be examined by radiography and by liquid penetrant examination in accordance with paragraph 3.8.3.1. Where the depth of repairs is less than 20% of the section thickness or 25 mm (1 in.), whichever are the lesser, and where the repaired section cannot be radiographed effectively, the first layer of each 6 mm (3/4 in.) thickness of deposited weld metal and the finished weld surface shall be examined, as indicated previously, by liquid penetrant examination. The finished surface examination shall be made after any heat treating operation that is applied to the casting. Weld repairs resulting from ultrasonic examination shall be examined by ultrasonic methods.

c) When repair welding is done after heat treatment of the casting, the casting shall be postweld heat treated after repair welding.

d) All welding shall be performed using welding procedures qualified in accordance with Section IX. The procedure qualifications shall be performed on test specimens of cast material of the same specification and subject to the same heat treatments before and after welding as will be applied to the work. All welders and welding operators performing this welding shall be qualified in accordance with Section IX.

e) The location and extent of the weld repairs together with the repair procedure and examination results shall be recorded and transmitted as part of the certification.

3.9 Supplemental Requirements for Hubs Machined From Plate

3.9.1 General

The supplemental requirements of paragraph 3.9 are required for plate materials that are used in the fabrication of hubs for tubesheets, lap joint stub ends, and flat heads machined from plate when the hub length is in the through thickness direction of the plate.

3.9.2 Material Requirements

3.9.2.1 Plate shall be manufactured by a process that produces material having through thickness properties which are at least equal to those specified in the material specification. Such plate can be but is not limited to that produced by methods such as electroslag (ESR) and vacuum arc re-melt (VAR). The plate must be tested and examined in accordance with the requirements of the material specification and the additional requirements specified in the following paragraphs.

3.9.2.2 Test specimens, in addition to those required by the material specifications, shall be taken in a direction parallel to the axis of the hub and as close to the hub as practical, as shown in Figure 3.2. At least two tensile test specimens shall be taken from the plate in the proximity of the hub, with one specimen taken from the center third of the plate width as rolled, and the second specimen taken at 90^{o} around the circumference from the other specimen. Both specimens shall meet the mechanical property requirements of the material specification. For carbon and low alloy steels, the reduction of area shall not be less than 30%; for those materials for which the material specification requires a reduction of area value greater than 30%, the higher value shall be met.

3.9.2.3 Subsize test specimens conforming to the requirements of SA-370, Figure 5 may be used if necessary, in which case the value for percent elongation in 50 mm (2 in.), required by the material specification, shall apply to the gage length specified in SA-370, Figure 5.

3.9.2.4 Tension test specimen locations are shown in Figure 3.2.

3.9.3 Examination Requirements

3.9.3.1 After machining the part, regardless of thickness, shall be ultrasonically examined by the straight beam technique in accordance with SA-388. The examination shall be in two directions approximately at right angles, i.e., from the cylindrical or flat rectangular surfaces of the hub and in the axial direction of the hub. The part shall be unacceptable if

a) The examination results show one or more indications accompanied by loss of back reflection larger than 60% of the reference back reflection and,

b) The examination results show indications larger than 40% of the reference back reflection when

accompanied by a 40% loss of back reflection.

3.9.3.2 Before welding the hub of the tubesheet flange or flat head to the adjacent shell, the hub shall be examined by magnetic particle or liquid penetrant methods in accordance with Part 7.

3.9.3.3 After welding, the weld and the area of the hub for at least 13 mm (1/2 in.) from the edge of the weld shall be 100% radiographed in accordance with Part 7. As an alternate, the weld and hub area adjacent to the weld may be ultrasonically examined in accordance with Part 7.

3.9.4 Data Reports and Marking

Whenever the provisions of this supplemental requirement are used, they shall be indicated on the Data Report. Special markings are not required.

3.10 Material Test Requirements

3.10.1 General

Material tests required by this Division shall be performed in accordance with paragraph 3.10.

3.10.2 Requirements for Sample Test Coupons

3.10.2.1 Heat Treatment

Heat treatment as used in this Division shall include all thermal treatments during fabrication at 480°C (900°F) and above.

3.10.2.2 Provisions of Sample Test Coupons

When material is subjected to heat treatment during fabrication, the test specimens required by this Division shall be obtained from sample coupons which have been heat treated in the same manner as the material, including such heat treatments as were applied by the material producer before shipment. The required tests may be performed by the material producer or the fabricator.

3.10.2.3 Heat Treating of Sample Test Coupons

a) The material used in the vessel shall be represented by test specimens that have been subjected to the same manner of heat treatment, including postweld heat treatment. The kind and number of tests and test results shall be as required by the material specification. The vessel Manufacturer shall specify the temperature, time, and cooling rates to which the material will be subject during fabrication. Material from which the specimens are prepared shall be heated at the specified temperature within the tolerance established by the manufacturer for use in actual fabrication. The total time at temperature shall be within at least 80% of the total time at temperature during actual heat treatment of the product and may be performed in a single cycle. Simulation of postweld heat treatment may be applied to the test specimen blanks.
b) Heat treatment of material is not intended to include such local heating as flame or arc cutting, preheating, welding, or heating below the critical range of tubing or pipe for bending or sizing.

3.10.3 Exemptions from Requirement of Sample Test Coupons

3.10.3.1 Standard Pressure Parts

An exception to the requirements of paragraphs 3.10.2.2 and 3.10.2.3 shall apply to standard items such as welded pipe or tubes and as described in paragraph 3.2.8. These may be subjected to postweld heat treatment with the vessel or vessel part without the same treatment being required of the test specimens. This exception shall not apply to castings that are specially designed or to cast wrought fittings.

3.10.3.2 For Materials When PWHT to Table 6.17

Materials listed in QW/QB-422 as P-No. 1 Group 3 and P-No. 3, Groups 1 and 2 that are certified in accordance with paragraphs 3.10.2.2 and 3.10.2.3 from test specimens subjected to the PWHT requirements of Table 6.10 need not be recertified if subjected to the alternative PWHT conditions permitted in Table 6.17.

3.10.3.3 Re-Austenitized Materials

All thermal treatments which precede a thermal treatment that fully austenitizes the material need not be accounted for by the specimen heat treatments provided the austenitizing temperature is at least as high as any of the preceding thermal treatments.

3.10.4 Procedure for Obtaining Test Specimens and Coupons

3.10.4.1 Plates

a) Unless otherwise specified, test specimens shall be taken in accordance with the requirements of the applicable material specification, except for the provisions in paragraphs (b), (c), and (d) below. Tension test specimens and Charpy V-notch specimens shall be orientated in the direction perpendicular to the final direction of the plate rolling.
b) When the plate is heat treated with a cooling rate faster than still-air cooling from the austenitizing temperature, the specimens shall be taken in accordance with requirements of applicable material specifications and 1-t from any heat treated edge, where t is the nominal thickness of the material.
c) Where a separate test coupon is used to represent the vessel material, it shall be of sufficient size to ensure that the cooling rate of the region from which the test specimens are removed represents the cooling rate of the material at least ¼-t deep and 1-t from any edge of the product. Unless cooling rates applicable to the bulk pieces or product are simulated in accordance with paragraph 3.10.5, the dimensions of the coupon shall be not less than 3-t x 3-t x 1-t, where t is the nominal thickness of the material.
d) When flat heads, tubesheets, and flanges with integral hubs for butt welding are to be machined from plate, additional specimens shall be taken in the locations as shown in Figure 3.2.

3.10.4.2 Forgings

a) Test specimens shall be taken in accordance with the applicable material specification, except for the provisions in paragraphs (b), (c), and (d) below.
b) When the forging is heat treated with a cooling rate faster than still-air cooling from the austenitizing temperature the specimens shall be taken at least ¼-t of the maximum heat treated thickness from one surface and 1t from a second surface. This is normally referred to as ¼-t x 1-t, where t is the maximum heat treated thickness. A thermal buffer may be used to achieve these conditions unless cooling rates applicable to the bulk forgings are simulated in accordance with paragraph 3.10.5.
c) For thick and complex forgings, such as contour nozzles, thick tubesheets, flanges, and other complex forgings that are contour shaped or machined to essentially the finished product configuration prior to heat treatment, the registered engineer who prepares the Design Report shall designate the surfaces of the finished product subject to high tensile stresses in service. Test specimens for these products shall be removed from prolongations or other stock provided on the product. The coupons shall be removed so that the specimens shall have their longitudinal axes at a distance below the nearest heat treated surface equivalent at least to the greatest distance that the indicated high tensile stress surface will be from the nearest surface during heat treatment and with the mid-length of the specimen at a minimum of twice this distance from the second heat treated surface. In any case, the longitudinal axes of the specimens shall not be nearer that 19 mm (3/4 in.) to any heat treated surface, and the mid-length of the specimens shall be at least 40 mm (1-1/2 in.) from any second heat treated surface.
d) With prior approval of the vessel Manufacturer, test specimens for flat ring and simple ring forgings may be taken from a separately forged piece under the following conditions.
    1) The separate test forging shall be of the same heat of material and shall be subjected to substantially the same reduction and working as the production forgings it represents.
    2) The separate test forging shall be heat treated in the same furnace charge and under the same conditions as the production forgings.
    3) The separate test forging shall be of the same nominal thickness as the production forgings. Test specimen material shall be removed as required in paragraphs 3.10.4.2.a) and b).

3.10.4.3 Tubular Products

Specimens shall be taken in accordance with the requirements of the applicable material specification.

3.10.4.4 Bars and Bolting Materials

a) Test specimens shall be taken in accordance with the requirements of the applicable material specification, except for the provisions of paragraph (b) below.
b) Test specimens shall be at least ¼-t from the outside or rolled surface and with the end of the specimen no closer than one diameter or thickness from the heat treated end.
c) For bolting, the specimens shall be taken in conformance with the applicable material specification and with the end of the specimen no closer than one diameter or thickness from a heat treated end.

3.10.4.5 Castings

a) The conventional ASTM separately cast test coupon meets the intent of paragraph 3.10.5 where normalizing or accelerated cooling heat treatments are employed on castings having a maximum thickness of less than 50 mm (2 in.).
b) For castings having a thickness of 50 mm (2 in.) and over, the specimens shall be taken from the casting (or the extension of it) at least ¼-t of the maximum heat treated thickness from one surface and 1-t from a second surface. A thermal buffer may be used.
c) For massive castings that are cast or machined to essentially the finished product configuration prior to heat treatment, the registered engineer who prepares the Design Report shall designate the surfaces of the finished product subject to high tensile stresses in service. Test specimens for these products shall be removed from prolongations or other stock provided on the product. The specimen shall be removed at a distance below the nearest heat treated surface equivalent at least to the greatest distance that the indicated high tensile stress surface will be from the nearest surface during heat treatment; the location shall also be a minimum of twice this distance from a second heat treated surface. In any case, specimen removal shall not be nearer than 19 mm (3/4 in.) to a heat treated surface and 38 mm (1-1/2 in.) to a second heat treated surface.
d) With prior approval of the vessel Manufacturer, test specimens for flat ring and simple ring forgings may be taken from a separately cast test coupon under the following conditions.
    1) The separate test coupon shall be of the same heat of material and shall be subjected to substantially the same casting practices as the production casting it represents.
    2) The separate test coupon shall be heat treated in the same furnace charge and under the same conditions as the production casting, unless cooling rates applicable to bulk castings are simulated in accordance with paragraph 3.10.5.
    3) The separate test coupon shall be of the same nominal thickness as the production casting. Test specimen material shall be removed from the region midway between mid-thickness and the surface and shall not be nearer than on thickness to a second surface.

3.10.5 Procedure for Heat Treating Test Specimens from Ferrous Materials

3.10.5.1 General requirements for heat treating of sample test coupons are covered in paragraph 3.10.2.3.

3.10.5.2 When ferritic steel products are subjected to normalizing or accelerated cooling from the austenitizing temperature, the test specimens representing those products shall be cooled at a rate similar to and no faster than the main body of the product except in the case of certain forgings and castings (see paragraphs 3.10.4.2.c and 3.10.4.5.c). This rule shall apply for specimens taken directly from the product as well as those taken from separate test coupons representing the product. The following general techniques may be applied to all product forms or test coupons representing the product.

a) Any procedure may be applied which can be demonstrated to produce a cooling rate in the test specimen that matches the cooling rate of the main body of the product at the region midway between mid-thickness and surface (¼-t) and no nearer any heat treated edge than a distance equal to the nominal thickness being cooled (t) within 14°C (25°F) and 20 seconds at all temperatures after cooling begins from the austenitizing temperature.
b) Faster cooling rates at product edges may be compensated for by:
    1) Taking the test specimens at least 1-t from a quenched edge where t equals the product thickness.
    2) Attaching a steel pad at least 1-t wide by a partial penetration weld to the product edge where specimen are to be removed.

3) Using thermal buffers or insulation at the product edge where specimens are to be removed.

c) If cooling rate data for the product and cooling rate device control devices for the test specimens are available, the test specimens may be heat treated in the device to represent the product provided that the provisions of paragraph 3.10.5.2.a) are met.

d) When the material is clad or weld deposit overlayed by the product prior to normalizing or accelerated cooling from the austenitizing temperature, the full thickness samples shall be clad or the weld deposit overlayed before such heat treatments.

3.10.6 Test Coupon Heat Treatment for Nonferrous Materials

3.10.6.1 Fabrication heat treatments of nonferrous material are normally not necessary. If heat treatment is performed, it shall be by agreement between the user and the vessel Manufacturer.

3.10.6.2 Materials where the mechanical properties are affected by fabrication heat treatments shall be represented by test specimens that have been subjected to the simulated fabrication heat treatments. The vessel Manufacturer shall specify the pertinent fabrication heat treatment parameters to the material manufacturer.

3.10.6.3 The requirements of paragraph 3.10.6.2 above exclude annealing and stress relieving.

3.11 Material Toughness Requirements

3.11.1 General

3.11.1.1 Charpy V-notch impact tests shall be made for materials used for shells, heads, nozzles, and other pressure containing parts, as well as for the structural members essential to structural integrity of the vessel, unless exempted by the rules of paragraph 3.11.

a) Toughness requirements for materials listed in Table 3.A.1 (carbon and low alloy steel materials except bolting materials) are given in paragraph 3.11.2.

b) Toughness requirements for materials listed in Table 3.A.2 (quenched and tempered steels with enhanced tensile properties) are given in paragraph 3.11.3.

c) Toughness requirements for materials listed in Table 3.A.3 (high alloy steels except bolting materials) are given in paragraph 3.11.4.

d) Toughness requirements for materials listed in Table 3.A.4 through 3.A.7 (nonferrous alloys) are given in paragraph 3.11.5.

e) Toughness requirements for all bolting materials are given in paragraph 3.11.6.

3.11.1.2 Toughness testing procedures and requirements for impact testing of welds and vessel test plates of ferrous materials are given in paragraphs 3.11.7 and 3.11.8, respectively.

3.11.1.3 Throughout paragraph 3.11, reference is made to the Minimum Design Metal Temperature (MDMT). The MDMT is part of the design basis of the vessel and is defined in paragraph 4.1.5.2.e. The rules in paragraph 3.11 are used to establish an acceptable MDMT for the material based on the materials of construction, product form, wall thickness, stress state, and heat treatment.

3.11.2 Carbon and Low Alloy Steels Except Bolting

3.11.2.1 Toughness Requirements for Carbon and Low Alloy Steels

a) Impact tests shall be performed on carbon and low alloy materials listed in Table 3.A.1 for all combinations of materials and MDMTs except as exempted by paragraph 3.11.2.3, 3.11.2.4, 3.11.2.5, or 3.11.2.8.

b) When impact testing is necessary, the following toughness values are required.

   1) If the specified minimum tensile strength is less than 655 MPA (95 ksi), then the required minimum energy requirement for all specimen sizes shall be that shown in Figure 3.3 and Figure 3.4 for vessel parts not subject to postweld heat treatment (PWHT) and vessel parts subject to PWHT,

respectively, multiplied by the ratio of the actual specimen width along the notch to the width of a full-size specimen, except as otherwise provided in paragraph 3.11.7.2.b.

2) If the specified minimum tensile strength is greater than or equal to 655 MPa (95 ksi), then the minimum lateral expansion (see Figure 3.5) opposite the notch for all specimen sizes shall not be less than the values shown in Figure 3.6.

3.11.2.2 Required Impact Testing Based On the MDMT, Thickness, and Yield Strength

a) If the governing thickness (see paragraph 3.11.2.3.b) at any welded joint or of any non-welded part exceeds 100 mm (4 in.) and the MDMT is colder than 43°C (110°F), then impact testing is required.

b) Unless specifically exempted in Figure 3.7 (for parts not subject to PWHT) or Figure 3.8 (for parts subject to PWHT and for non-welded parts), materials having a specified minimum yield strength greater than 450 MPa (65 ksi) shall be impact tested.

3.11.2.3 Exemption from Impact Testing Based On the MDMT, Thickness, and Material Specification

a) Figure 3.7 for parts not subject to PWHT or Figure 3.8 for parts subject to PWHT shall be used to establish impact testing exemptions based on the impact test exemption curve for the subject material specification and grade or class of the steel, MDMT, and governing thickness of a welded part. If an MDMT and thickness combination for the subject material is on or above the applicable impact test exemption curve in Figure 3.7 or Figure 3.8, then impact testing is not required except as required by paragraph 3.11.8 for weld metal and heat affected zones.

b) The governing thickness, t_g, of a welded part is determined using the following criteria. Examples of the governing thickness for some typical vessel details are shown in Figures 3.9, 3.10, and 3.11.

   1) For all product forms except castings:

      i) For butt joints except those in flat heads and tubesheets, the nominal thickness of the thickest welded joint [see Figure 3.9, sketch (a)],

      ii) For corner, fillet, or lap welded joints, including attachments as defined in paragraph 3.11.1.1, the thinner of the two parts joined,

      iii) For flat heads or tubesheets, the larger of paragraph (ii) above or the flat component thickness divided by 4.

   2) The governing thickness of a casting shall be its largest nominal thickness.

   3) The governing thickness of flat nonwelded parts, such as bolted flanges, tubesheets, and flat heads, is the flat component thickness divided by 4.

   4) The governing thickness of a nonwelded dished head is the greater of the flat flange thickness divided by 4 or the minimum thickness of the dished portion.

c) Components such as shells, heads, nozzles, manways, reinforcing pads, stiffening rings, flanges, tubesheets, flat cover plates, backing strips, and attachments that are essential to the structural integrity of the vessel when welded to pressure retaining components shall be treated as separate components. Each component shall be evaluated for impact test requirements based on its individual material classification, governing thickness (see paragraph 3.11.2.3.b), and the MDMT. For welded assemblies comprised of more than two components (e.g., nozzle-to-shell joint with reinforcing pad), the governing thickness and permissible MDMT of each of the individual welded joints of the assembly shall be determined, and the warmest MDMT shall be used as the permissible MDMT of the welded assembly.

d) Figure 3.7 limits the maximum nominal governing thickness for welded parts not subject to postweld heat treatment to 38 mm (1-1/2 in.). Some vessels may have welded non-postweld heat treated pressure parts whose thickness exceeds the nominal governing thickness of 38 mm (1-1/2 in.). Examples of such welded and non-post heat treated pressure parts are thick tubesheets, flat heads, and thick insert plates (with beveled edges) with nozzles or load carrying structural attachments. Such welded non-postweld heat treated pressure parts shall be impact tested and shall meet the impact test requirements of this Division.

e) Impact testing is not required for materials with a thickness of 2.5 mm (0.099 in.) and thinner, but such exempted materials shall not be used at design metal temperatures colder than -48°C (-55°F). For components made from DN 100 (NPS 4) pipe or smaller and for equivalent size of tubes of P-No. 1 materials, the following exemptions from impact testing are also permitted as a function of the specified

minimum yield strength (SMYS) of the material for metal temperatures of -104°C (-155°F) and warmer:

1) For SMYS between 140 - 240 MPa (20 -35 ksi), inclusive, the thickness exemption for impact testing is 6 mm (1/4 in.).
2) For SMYS between 250 - 310 MPa (36-45 ksi), inclusive, the thickness exemption for impact testing is 3.2 mm (1/8 in.).
3) For SMYS higher than 315 MPa (46 ksi), inclusive, the thickness exemption for impact testing is 2.5 mm (0.099 in.).

f) Note that the rules in this paragraph for the exemption of impact testing do not provide assurance that all test results for these materials will satisfy the impact test acceptance criteria of paragraph 3.11.2.1.b.

3.11.2.4 Exemption from Impact Testing Based On Material Specification and Product Form

a) Impact testing is not required for the ferritic steel flanges shown below when supplied in heat treated condition (normalized, normalized and tempered, or quenched and tempered) and used at design temperatures no colder than -29°C (-20°F) and no colder than -18°C (0°F) when supplied in the as-forged condition:
   1) ASME B16.5 flanges,
   2) ASME B16.47 flanges,
   3) Long weld neck flanges, defined as forged nozzles that meet the dimensional requirements of a flanged fitting given in ASME B16.5 but have a straight hub/neck. The neck inside diameter shall not be less than the nominal size of the flange, and the outside diameter of the neck and any nozzle reinforcement shall not exceed the diameter of the hub as specified in ASME B16.5.

b) Materials produced and impact tested in accordance with the requirements of the specifications shown below are exempt from impact testing by the rules of this Division at MDMTs not more than 3 ºC (5 ºF) colder than the test temperature required by the specification.
   1) SA-320
   2) SA-333
   3) SA-334
   4) SA-350
   5) SA-352
   6) SA-420
   7) SA-437
   8) SA-508 Grade 5 Class 2
   9) SA-540 except for materials produced under Table 2, note 4 in this specification
   10) SA-723
   11) SA-765

3.11.2.5 Exemption from Impact Testing Based On Design Stress Values

a) A colder MDMT for a component than that derived from paragraph 3.11.2.3 may be determined in accordance with the procedure outlined below.
   1) STEP 1 – For the welded part under consideration, determine the nominal thickness of the part, t_n, and the required governing thickness of the part, t_g, using paragraph 3.11.2.3.b.
   2) STEP 2 – Determine the applicable material toughness curve to be used in Figure 3.7 for parts not subject to PWHT or Figure 3.8 for parts subject to PWHT. A listing of material assignments to the toughness curves is provided in the Material Assignment Table for Figure 3.8M.
   3) STEP 3 – Determine the MDMT from Figure 3.7 for parts not subject to PWHT or Figure 3.8 for parts subject to PWHT based on the applicable toughness curve and the governing thickness, t_g.

4) STEP 4 – Based on the design loading conditions at the MDMT, determine the stress ratio, R_{ts}, using one of the equations shown below. Note that this ratio can be computed in terms of required design thickness and nominal thickness, applied stress and allowable design stress, or applied pressure and maximum allowable working pressure based on the design rules in this Division or ASME/ANSI pressure-temperature ratings.

$$R_{ts} = \frac{t_r E^*}{t_n - CA} \qquad (Thickness\ Basis) \tag{3.1}$$

$$R_{ts} = \frac{S^* E^*}{SE} \qquad (Stress\ Basis) \tag{3.2}$$

$$R_{ts} = \frac{P_a}{P_{rating}} \qquad (Pressure-Temperature\ Rating\ Basis) \tag{3.3}$$

5) STEP 5 – Determine the final value of the MDMT and evaluate results

i) If the computed value of the R_{ts} ratio from STEP 4 is less than or equal to the 0.24, then set the MDMT to $-104^oC\ (-155^oF)$. Impact testing is not required unless a lower MDMT is required.

ii) If the computed value of the R_{ts} ratio from STEP 4 is greater than 0.24, then determine the temperature reduction, T_R. If the specified minimum yield strength is less than or equal to 450 MPa (65 ksi), then determine T_R from Figure 3.12 for parts not subject to PWHT or Figure 3.13 for parts subject to PWHT based on the R_{ts} ratio from STEP 4. If the specified minimum yield strength is greater than 450 MPa (65 ksi), then determine the temperature reduction, T_R from Equation (3.4). The final computed value of the MDMT is determined using Equation (3.5). The reduction in the MDMT given by Equation (3.5) shall not exceed 55°C (100°F). Impact testing is not required if the specified MDMT is warmer than the computed MDMT. However, if the specified or computed MDMT are colder than -48°C (-55°F), impact testing is required.

$$T_R = \frac{\begin{pmatrix} -27.20656 - 76.98828R_{ts} + \\ 103.0922R_{ts}^2 + 7.433649(10)^{-3}S_y \end{pmatrix}}{\begin{pmatrix} 1 - 1.986738R_{ts} - 1.758474(10)^{-2}S_y + \\ 6.479033(10)^{-5}S_y^2 \end{pmatrix}} \qquad ({}^oF,\ ksi) \tag{3.4}$$

$$MDMT = MDMT_{STEP3} - T_R \tag{3.5}$$

b) The procedure in paragraph 3.11.2.5.a) above is repeated for each welded part, and the warmest MDMT of all welded parts is the MDMT for the vessel.

c) For a flange attached by welding, the procedure in paragraph 3.11.2.5.a) above can be used by determining the temperature reduction as determined for the neck or shell to which the flange is attached. The bolt-up condition need not be considered when determining the temperature reduction for flanges.

d) Figure 3.12 for parts not subject to PWHT or Figure 3.13 for parts subject to PWHT may be used for components not stressed in primary membrane tensile stress, such as flat heads, covers, tubesheets, and flanges (including bolts and nuts). The MDMT shall not be colder than the impact test temperature less the allowable temperature reduction as determined from Figures 3.12 and 3.13. The ratio used in

paragraph 3.11.2.5.a), STEP 4, shall be the ratio of the maximum design pressure at the MDMT to the maximum allowable pressure (MAP) of the component at the MDMT.

3.11.2.6 Adjusting the MDMT for Impact Tested Materials

a) For components that are impact tested, the components may be used at a MDMT colder than the impact test temperature, provided the stress ratio in Figure 3.12 for parts not subject to PWHT or Figure 3.13 for parts subject to PWHT is less than one and the MDMT is not colder than -104°C (-155°F). For such components, the MDMT shall not be colder than the impact test temperature less the allowable temperature reduction as determined from paragraph 3.11.2.5 (i.e., the starting point for the MDMT calculation in STEP 3 is the impact test temperature). (See paragraph 3.11.2.4.b).
b) One common usage of the exemptions in paragraphs 3.11.2.5 and 3.11.2.6 will be for vessels in which the pressure is dependent on the vapor pressure of the contents (e.g., vessels in refrigeration plants and those subject to low seasonal atmospheric temperatures). For such services, the primary thickness calculations normally will be made for the maximum design pressure coincident with the maximum temperature expected above the line in Figure 3.3 (for as-welded parts) or Figure 3.4 (for stress relieved parts) for the applicable group of materials, using the appropriate stress intensity values from Annex 3.A. Thickness calculations then will be made for the maximum coincident pressure expected below the line in Figure 3.3 or 3.4 (as applicable) for the applicable group of materials using the reduced design stress intensity value(s). The greater of the thicknesses so calculated shall be used. Comparison of pressure ratios to stress ratios may suffice when loadings not caused by pressure are insignificant.

3.11.2.7 Vessel or Components Operating Below the MDMT

Vessels or components may be operated at temperatures colder than the MDMT stamped on the name- plate if:

a) The provisions of paragraph 3.11.2 are met when using the reduced (colder) operating temperature as the MDMT, but in no case shall the operating temperature be colder than -104°C (-155°F); or
b) For vessels or components whose thicknesses are based on pressure loading only, the coincident operating temperature may be as cold as the MDMT stamped on the nameplate less the allowable temperature reduction as determined from paragraph 3.11.2.5. The ratio used in Step 4 of the procedure in paragraph 3.11.2.5 shall be the ratio of maximum pressure at the coincident operating temperature to the design pressure of the vessel at the stamped MDMT, but in no case shall the operating temperature be colder than -104°C (-155°F).

3.11.2.8 Establishment of the MDMT Using a Fracture Mechanics Methodology

a) In lieu of the procedures in paragraphs 3.11.2.1 thru 3.11.2.7, the MDMT may be established using a fracture mechanics approach. The fracture mechanics procedures shall be in accordance with API 579-1/ASME FFS, Part 9, Level 2 or Level 3.
b) The assessment used to determine the MDMT shall include a systematic evaluation of all factors that control the susceptibility to brittle fracture, e.g. stresses from the applied loadings including thermal stresses, flaw size, fracture toughness of the base metal and welded joints, heat treatment, and the loading rate.
c) The reference flaw size used in the fracture mechanics evaluation shall be a surface flaw with a depth of $a = \min\left[t/4,\ 25\ mm\ (1\ in.)\right]$ and a length of $2c = 6a$ where t is the thickness of the plate containing the reference flaw. If approved by the user, an alternative reference flaw size may be used based on the weld joint geometry and the NDE that will be used and demonstrated for qualification of the vessel (see Part 7).
d) The material fracture toughness shall be established using the exemption curve for the material (see Notes to Figures 3.7 and 3.8) and MPC Charpy impact energy correlation described in API 579-1/ASME FFS-1, Appendix F, paragraph F.4. If approved by the user, an alternative material fracture toughness may be used based on fracture toughness test results.
e) The MDMT established using a fracture mechanics approach shall not be colder than that given in paragraph 3.11.2.3.e.

3.11.2.9 Postweld Heat Treatment Requirements for Materials in Low Temperature Service

a) If the MDMT is colder than -48°C (-55°F) and the stress ratio in Figure 3.12 for parts not subject to

PWHT or Figure 3.13 for parts subject to PWHT is greater than or equal to 0.24, then welded joints shall be subject to PWHT in accordance with the requirements of paragraph 6.4.2.

b) The requirement in paragraph (a) above does not apply to the welded joints listed in paragraphs (b)(1) and (b)(2) below in vessel or vessel parts fabricated of P-No. 1 materials that are impact tested at the MDMT or colder in accordance with paragraph 3.11.2.1. The minimum average energy requirement for base metal, weld metal, and heat affected zones shall be 41J (30 ft-lb) instead of the values shown in Figure 3.3 for parts not subject to PWHT or Figure 3.4 for parts subject to PWHT.
    1) Type 1 Category A and B joints, not including cone-to-cylinder junctions, that have been 100% radiographed. Category A and B joints attaching sections of unequal thickness shall have a transition with a slope not exceeding 3:1.
    2) Fillet welds having leg dimensions not exceeding 10 mm (3/8 in.) attaching lightly loaded attachments, provided the attachment material and the attachment weld meet the requirements of paragraph 3.11.2 and paragraph 3.11.8. Lightly loaded attachments, for this application, are defined as attachments in which the stress in the attachment weld does not exceed 25% of the allowable stress. All such welds shall be examined by liquid penetrant or magnetic particle examination in accordance with Part 7 of this Division.

3.11.2.10 Impact Tests of Welding Procedures

a) For welded construction, the Welding Procedure Qualification shall include impact tests of welds and heat affected zones when required by the following provisions.
b) Welds made with filler metal shall be impact tested in accordance with paragraph 3.11.2.1 when any of the following apply:
    1) When either base metal is required to be impact tested by the rules of this Division;
    2) When joining base metals from Figure 3.7 or Figure 3.8, Curves C or D, or metals exempted from impact testing by paragraph 3.11.2.4.b, and the MDMT is colder than -29°C (-20°F) but not colder than -48°C (-55°F), unless the welding consumables that have been classified by impact tests at a temperature not warmer than the MDMT by the applicable SFA specification are used;
    3) When joining base metals exempt from impact testing by paragraphs 3.11.2.3, 3.11.2.4 and 3.11.2.5 when the MDMT is colder than -48°C (-55°F).
c) Welds in materials made without the use of filler metal shall be impact tested when the thickness of the weld exceeds 13 mm (1/2 in.) for all MDMTs or when the thickness at the weld exceeds 8 mm (5/16 in.) and the MDMT is colder than 10°C (50°F). This requirement does not apply to welds made as part of the material specification.
d) Weld heat affected zones produced with or without the addition of filler metal shall be impact tested whenever any of the following apply:
    1) When the base metal is required to be impact tested by the rules of this Division;
    2) When the welds have any individual weld pass exceeding 13 mm (0.5 in.) in thickness, and the MDMT is colder than 21°C (70°F);
    3) When joining base metals exempt from testing by paragraph 3.11.2.4.b when the MDMT is colder than -48°C (-55°F).
e) Vessel production impact tests in accordance with paragraph 3.11.8.4 may be waived for any of the following:
    1) Weld metals joining steels exempted from impact testing by paragraphs 3.11.2.3, 3.11.2.4 and 3.11.2.5 for minimum MDMTs of -29°C (-20°F) or warmer;
    2) Weld metals defined in paragraph 3.11.2.10.b.2 and paragraph 3.11.2.10.b.3;
    3) Heat affected zones in steels exempted from impact testing by paragraphs 3.11.2.3, 3.11.2.4 and 3.11.2.5 except when paragraph 3.11.2.10.d.3 applies.

3.11.3 Quenched and Tempered Steels

3.11.3.1 Toughness Requirements for Quenched and Tempered Ferritic Steels

a) All quenched and tempered steels listed in Table 3.A.2 shall be subject to Charpy V-notch testing.
b) Impact tests shall be conducted at a temperature not warmer than the MDMT determined in Part 4,

paragraph 4.1.5.2.d. However, in no case shall the MDMT be warmer than 0°C (32°F).

c) Materials may be used at temperatures colder than the MDMT as permitted below.

   1) When the stress ratio defined in Figure 3.12 for parts not subject to PWHT or Figure 3.13 for parts subject to PWHT is 0.24 or less, the corresponding MDMT shall not be colder than -104°C (-155°F).

   2) When the stress ratio defined in Figure 3.12 for parts not subject to PWHT or Figure 3.13 for parts subject to PWHT is greater than 0.24, the corresponding MDMT shall not be colder than the impact test temperature less the allowable temperature reduction permitted in Figure 3.12 for parts not subject to PWHT or Figure 3.13 for parts subject to PWHT and shall in no case be colder than -104°C (-155°F).

3.11.3.2 Impact Testing

a) Preparation Of Test Specimens – All test specimens shall be prepared from the material in its final heat treated condition according to the requirements of paragraph 3.11.7.2.

b) Number of Impact Tests and Test Specimens – One Charpy V-notch impact test shall consist of three test specimens. For as-rolled plates, one Charpy V-notch test shall be made from each as-rolled plate. For heat treated plates (normalized, normalized and tempered, or quenched and tempered), one Charpy V-notch test shall be made from each plate-as-heat-treated. One Charpy V-notch test shall be made from each heat of bars, pipe, tubing, rolled sections, forged parts or castings included in any one heat treatment lot. The number of impact tests shall not be less than required by the material specification.

c) Locations and Orientation of Test Specimens – The location and orientation of the specimens shall be the same as required for Charpy type impact tests by paragraph 3.11.7.2 and paragraph 3.11.7.3 except that specimens from plates shall be transverse to the final direction of rolling and for forgings and pipe, transverse to the direction of major work (see Figure 3.14).

d) The minimum lateral expansion shall be in accordance with paragraph 3.11.2.1.b.

e) Retesting shall be in accordance with paragraph 3.11.7.6.

3.11.3.3 Drop-Weight Tests

a) When the MDMT is colder than -29°C (-20°F), drop-weight tests as defined by ASTM E 208, Conducting Drop-Weight Test to Determine Nil-Ductility Transition Temperature of Ferritic Steels, shall be made on all materials listed in Table 3.A.2, with the following exceptions:

   1) SA-522 for any MDMT;

   2) SA-353 and SA-553 when the temperature is not colder than -196°C (-320°F);

   3) SA-645 Grade A when the temperature is not colder than -170°C (-275°F).

b) Number of Tests for Plates – For plates 16 mm (5/8 in.) thick and greater, one drop-weight test (two specimens) shall be made for each plate in the as-heat-treated condition (see paragraph 3.11.3.2).

c) Number of Tests for Forgings and Castings – For forgings and castings of all thicknesses, one drop-weight test (two specimens) shall be made for each heat in anyone heat treatment lot. The sampling procedure shall comply with the requirements of ASTM E 208.

d) Required Test Results – Each of the two test specimens shall meet the "no-break" criterion, as defined by ASTM E 208, at the test temperature.

3.11.4 High Alloy Steels Except Bolting

3.11.4.1 Toughness Requirements for High Alloy Steels

a) Impact tests shall be performed on high alloy materials listed in Table 3.A.3 for all combinations of materials and MDMTs except as exempted by paragraph 3.11.4.3 or 3.11.4.5.

b) When impact testing is required, the minimum lateral expansion opposite the notch shall be 0.38 mm (0.015 in.) for MDMTs of -196°C (-320°F) and warmer. For MDMTs colder than this temperature, production welding processes shall be limited to shielded metal arc welding (SMAW), gas metal arc welding (GMAW), submerged arc welding (SAW), plasma arc welding (PAW), and gas tungsten arc welding (GTAW). Each heat, lot or batch of filler metal and filler metal/flux combination shall be pre-use tested as required by paragraph 3.11.4.5(d)(1) through (3). Exemption from pre-use testing as allowed by paragraphs 3.11.4.5(d)(4) and 3.11.4.5(d)(5) is not applicable. Notch toughness testing shall be performed as specified in paragraphs (1) or (2) below, as appropriate.

1) For Type 316L weld filler metal:
   i) Each heat of filler metal used in production shall have a Ferrite Number not greater than 5, as measured by a ferritescope or magna gauge calibrated in accordance with AWS A 4.2 or as determined by applying the chemical composition from the test weld to Figure 3.15.
   ii) Notch toughness testing of the base metal, weld metal, and heat affected zone (HAZ) shall be conducted using a test temperature of -196°C (-320°F).
   iii) Each of the three specimens from each test set shall have a lateral expansion opposite the notch not less than 0.53 mm (0.021 in.).
2) If using filler metals other than Type 316L, or when the base metal, weld metal, or heat affected zone are unable to meet the requirements of paragraph (1) above:
   i) Notch toughness testing shall be conducted at a test temperature not warmer than MDMT, using the ASTM E 1820 JIC method.
   ii) A set of two specimens shall be tested in the TL orientation with a resulting KIC (JIC) of not less than 132 MPa√m (120 ksi√in).
   iii) Each heat or lot of austenitic stainless steel filler metal used in production shall have a Ferrite Number not greater than the Ferrite Number detemined for the test weld.

3.11.4.2 Required Impact Tests When Thermal Treatments Are Performed

Impact tests are required at the colder of 21°C (70°F) or the MDMT whenever thermal treatments within the temperature ranges listed for the following materials are applied.

a) Austenitic stainless steels thermally treated between 480°C and 900°C (900°F and 1650°F), except for Types 304, 304L, 316, and 316L which are thermally treated at temperatures between 480°C and 705°C (900°F and 1300°F) are exempt from impact testing provided the MDMT is -29°C (-20°F) and warmer and vessel production impact tests of the thermally treated weld metal are performed for Category A and B joints.
b) Austenitic-ferritic duplex stainless steels thermally treated at temperatures between 315°C and 955°C (600°F and 1750°F).
c) Ferritic chromium stainless steels and martensitic chromium stainless steels thermally treated at temperatures between 425°C and 730°C (800°F and 1350°F).

Thermal treatments of materials do not include thermal cutting.

3.11.4.3 Exemptions from Impact Testing for Base Materials and Heat Affected Zones

Impact testing is not required for the following combinations of base metals and heat affected zones (if welded) and MDMT, except as modified in paragraph 3.11.4.2.

a) For austenitic chromium-nickel stainless steels as follows:
   1) Types 304, 304L, 316, 316L, 321, and 347 at MDMTs of -196°C (-320°F) and warmer;
   2) Those types not listed in paragraph (1) above and having a carbon content not exceeding 0.10% at MDMTs of -196°C (-320°F) and warmer. (The value of the carbon content may be specified by the purchaser, or must be within the limits of the material specification.);
   3) Those types having a carbon content exceeding 0.10% (the value of the carbon content may be as specified by the purchaser) at MDMTs of -48°C (-55°F) and warmer;
   4) For castings at MDMTs of -29°C (-20°F) and warmer.
b) For austenitic chromium-manganese-nickel stainless steels (200 series) as follows:
   1) Having a carbon content not exceeding 0.10% at MDMTs of -196°C (-320°F) and warmer;
   2) Having a carbon content exceeding 0.10% at MDMTs of -48°C (-55°F) and warmer;
   3) For castings at MDMTs of -29°C (-20°F) and warmer.
c) For the following steels in all product forms at MDMTs of -29°C (-20°F) and warmer:
   1) Austenitic-ferritic ferritic duplex steels with a nominal material thickness of 10 mm (3/8 in.) and thinner;
   2) Ferritic chromium stainless steels with a nominal material thickness of 3 mm (1/8 in.) and thinner;

3) Martensitic chromium stainless steels with a nominal material thickness of 6 mm (1/4 in.) and thinner.

d) Impact tests are not required where the maximum obtainable Charpy specimen has a width along the notch less than 2.5 mm (0.099 in).

e) Impact testing of materials is not required, except as modified by paragraph 3.11.4.2, when the coincident ratio of applied stress in tension to allowable tensile stress is less than 0.24. The applied stress is the stress from pressure and non-pressure loadings, including those listed in Table 4.1.1 which result in general primary membrane tensile stress.

3.11.4.4 Exemptions from Impact Testing for Welding Procedure Qualifications

For welding procedure qualifications, impact testing is not required for the following combinations of weld metals and MDMT except as modified by paragraph 3.11.4.2.

a) For austenitic chromium-nickel stainless steel base materials having a carbon content not exceeding 0.10%, welded without the addition of filler metal, at MDMTs of -104°C (-155°F) and warmer.

b) For austenitic weld metal:

   1) Having a carbon content not exceeding 0.10% and produced with filler metals conforming to SFA-5.4, SFA-5.9, SFA-5.11, SFA-5.14, and SFA-5.22 at MDMTs of -104°C (-155°F) and warmer;

   2) Having a carbon content exceeding 0.10% and produced with filler metals conforming, to SFA- 5.4, SFA-5.9, SFA-5.11, SFA-5.14, and SFA-5.22 at MDMTs of -48°C (-55°F) and warmer.

c) For the following weld metal, if the base metal of similar chemistry is exempt as stated in paragraph 3.11.4.3.c above, then the weld metal shall also be exempt at MDMTs of -29°C (-20°F) and warmer:

   1) Austenitic-ferritic duplex steels;

   2) Ferritic chromium stainless steels; and

   3) Martensitic chromium stainless steels.

3.11.4.5 Required Impact Testing for Austenitic Stainless Steel Welding Consumables with MDMT Colder Than -104 ºC (-155 ºF)

For production welds at MDMTs colder than -104 ºC (-155 ºF), all of the following conditions shall be satisfied:

a) The welding processes are limited to SMAW, SAW, GMAW, GTAW, and PAW;

b) The applicable Welding Procedure Specifications (WPSs) are supported by Procedure Qualification Records (PQRs) with impact testing in accordance with the requirements of paragraph 3.11.7 (using the acceptance criteria of paragraph 3.11.4.1) at the MDMT or colder, or when the applicable PQR is exempted from impact testing by other provisions of this Division;

c) The weld metal (produced with or without the addition of filler metal) has a carbon content not exceeding 0.10%;

d) The weld metal is produced by filler metal conforming to Section II, Part C, SFA-5.4, SFA-5.9, SFA-5.11, SFA-5.14, and SFA-5.22 as modified below.

   1) Each heat and/or lot of welding consumables to be used in production welding with the SMAW and GMAW processes shall be pre-use tested by conducting impact tests at the MDMT or colder. Test coupons shall be prepared in accordance with Section II, Part C, SFA-5.4, A9.3.5 uitilizing the WPS to be used in production welding. Acceptance criteria shall conform with paragraph 3.11.4.1.

   2) Each heat of filler metal and batch of flux combination to be used in procuction welding with the SAW process shall be pre-use tested by condicting impact tests at the MDMT or colder. Test coupons shall be prepared in accordance with Section II, Part C, SFA-5.4, A9.3.5 uitilizing the WPS to be used in production welding. Acceptance criteria shall conform with paragraph 3.11.4.1.

   3) Combining more than one welding process or more than one heat, lot, and/or batch of welding material into a single test coupon is unacceptable. Pre-use testing at the MDMT or colder may be conducted by the welding consumable manufacturer provided certified mill test reports are furnished with the consumables.

   4) The following filler metals may be used without pre-use testing of each heat, lot, and/or batch provided that the procedure qualification impact testing in accordance with paragraph 3.11.8 at the MDMT or colder is preformed using the same manufacturer brand and type filler metal: ENiCrFe-2;

ENiCrFe-3; ENiCrMo-3; ENiCrMo-4; ENiCrMo-6; ERNiCr-3; ERNiCrMo-3; ERNiCrMo-4; SFA-5.4, E310-15 or 16.

5) The following filler metals may be used without pre-use testing of each heat and/or lot provided that procedure qualification impact testing in accordance with paragraph 3.11.8 at the MDMT or colder is performed: ER308L, ER316L, and ER310 used with the GTAW or PAW processes.

3.11.4.6 Required Impact Testing for Vessel Production Test Plates

a) For welded construction, of duplex stainless steels, ferritic stainless steels and martensitic stainless steels, vessel production impact tests in accordance with paragraph 3.11.8.4 are required if the welding procedure qualification requires impact testing, unless otherwise exempted by the rules of this Division.

b) If the MDMT is colder than 196 ºC (-320 ºF), then vessel production impact tests or ASTM E 1820 J_{IC} tests shall be conducted in accordance with paragraph 3.11.4.1.

c) Vessel Production Impact Testing for Autogeneous Welds in Austenitic Stainless Steels – For autogenous welds (welded without filler metal) in austenitic stainless steels, vessel (production) impact tests are not required when all of the following conditions are satisfied:

1) The material is solution annealed after welding.

2) The MDMT is not colder than -196 ºC (-320 ºF).

d) Required Impact Testing for Austenitic Stainless Steel Welding Consumables With MDMT Colder than -104 ºC (-155 ºF) – For production welds at MDMTs colder -104°C (-155°F), all the of the following conditions shall be satisfied:

1) The welding processes are limited to SMAW, SAW, GMAW, GTAW, and PAW;

2) The applicable Welding Procedure Specifications (WPSs) are supported by Procedure Qualification Records (PQRs) with impact testing in accordance with the requirements of paragraph 3.11.7 (using the acceptance criteria of paragraph 3.11.4.1) at the MDMT or colder, or when the applicable PQR is exempted from impact testing by other provisions of this Division;

3) The weld metal (produced with or without the addition of filler metal) has a carbon content not exceeding 0.10%;

4) The weld metal is produced by filler metal conforming to Section II, Part C, SFA-5.4, SFA-5.9, SFA-5.11, SFA-5.14, and SFA-5.22 as modified below.

i) Each heat and/or lot of welding consumables to be used in production welding with the SMAW and GMAW processes shall be pre-use tested by conducting impact tests at the MDMT or colder. Test coupons shall be prepared in accordance with Section II, Part C, SFA-5.4, A9.l2. Acceptance criteria shall conform with paragraph 3.11.4.1.

ii) Each heat of filler metal and batch of flux combination to be used in production welding with the SAW process shall be pre-use tested by conducting impact tests at the MDMT or colder. Test coupons shall be prepared in accordance with Section II, Part C, SFA-5.4, A9.12. Acceptance criteria shall conform with paragraph 3.11.4.1.

iii) Combining more than one welding process or more than one heat, lot, and/or batch of welding material into a single test coupon is unacceptable. Pre-use testing at the MDMT or colder may be conducted by the welding consumable manufacturer provided certified mill test reports are furnished with the consumables.

iv) The following filler metals may be used without pre-use testing of each heat lot, and/or batch provided that procedure qualification impact testing in accordance with paragraph 3.11.8 at the MDMT or colder is performed using the same manufacturer brand and type filler metal: ENiCrFe-2; ENiCrFe-3; ENiCrMo-3; ENiCrMo-4; ENiCrMo-6; ERNiCr-3; ER-NiCrMo-3; ERNiCrMo-4; SFA-5.4, E310-15 or 16.

v) The following filler metals may be used without pre-use testing of each heat and/or lot provided that procedure qualification impact testing in accordance with paragraph 3.11.8 at the MDMT or colder is performed: ER308L, ER316L, and ER310 used with the GTAW, or PAW processes.

3.11.5 Non-Ferrous Alloys

3.11.5.1 Non-Ferrous materials listed in Tables 3.A.4 thru 3.A.7, together with deposited weld metal within the range of composition for material in that Table, do not undergo a marked drop in impact resistance at subzero temperature. Therefore, additional requirements are not specified for:

a) Wrought aluminum alloys when they are used at temperature down to -269°C (-452°F);
b) Copper and copper alloys, nickel and nickel alloys, and cast aluminum alloys when they are used at temperatures down to -198°C (-325°F); and
c) Titanium or zirconium and their alloys used at temperatures down to -59°C (-75°F).

3.11.5.2 The nonferrous materials listed in Tables 3.A.4 thru 3.A.7, may be used at lower temperatures than those specified herein and for other weld metal compositions provided the user satisfies himself by suitable test results such as determinations of tensile elongation and sharp-notch tensile strength (compared to unnotched tensile strength) that the material has suitable ductility at the design temperature.

3.11.6 Bolting Materials

3.11.6.1 Bolting Materials for Use With Flanges Designed To Paragraph 4.16

a) Impact tests are not required for bolting materials listed in Tables 3.4, 3.5, 3.6, and 3.7 when used at MDMTs equal to or warmer than those shown in these Tables.
b) Bolting materials to be used for colder temperatures than those shown in Tables 3.4 - 3.7 shall conform to SA-320, except that the toughness criterion shall be Charpy V-notch with acceptance criteria in accordance with paragraph 3.11.2 or 3.11.4, as applicable.

3.11.6.2 Bolting Materials for Use with Flanges Designed To Part 5 of This Division

Impact testing is required for the ferrous bolting materials listed in Table 3.A.11 for use with flanges designed in accordance with Part 5 of this Division. The average for three Charpy V -notch impact specimens shall be at least 41J (30 ft-lb), with the minimum value for any individual specimen not les than 34J (25 ft-lb).

3.11.7 Toughness Testing Procedures

3.11.7.1 Test Procedures

a) Impact test procedures and apparatus shall conform to the applicable paragraphs of SA-370 or ISO148 (Parts 1, 2, and 3).
b) The impact test temperature shall not be warmer than the MDMT (see paragraph 4.1.5.2.e).

3.11.7.2 Test Specimens

a) Each set of impact tests shall consist of three specimens.
b) The impact test specimens shall be of the Charpy V-notch type and shall conform in all respects to the specimen requirements of SA-370 (for Type A specimens). The standard full-size (10 mm X 10 mm specimen, when obtainable, shall be used, except that for materials that normally have absorbed energy in excess of 244 J (180 ft-lb) when tested using full size specimens at the specified testing temperature, subsize (10 mm x 6.7 mm) specimens may be used in lieu of full-size specimens. However, when this option is used, the acceptance value shall be 102 J (75 ft-lb) minimum for each specimen.
c) For material from which full-size specimens cannot be obtained, either due to the material shape or thickness, the specimens shall be either the largest possible subsize specimen obtainable or specimens of full material thickness which may be machined to remove surface irregularities [the test temperature criteria of paragraph 3.11.7.5 shall apply for carbon and low alloy materials having a specified minimum tensile strength less than 655 MPa (95 ksi) when the width along the notch is less than 80% of the material thickness]. Alternatively, such material may be reduced in thickness to produce the largest possible Charpy subsize specimen. Toughness tests are not required where the maximum obtainable Charpy specimen has a width along the notch less than 2.5 mm (0.099 in.), but carbon steels too thin to impact test shall not be used for design temperatures colder than -48°C (-55°F), subject to the exemptions provided by paragraph 3.11.2.9.

3.11.7.3 Product Forms

a) Impact test specimens of each product form shall be located and oriented in accordance with the requirements of paragraph 3.10.4.

b) The manufacturer of small parts, either cast or forged, may certify a lot of not more than 20 duplicate parts by reporting the results of one set of impact specimens taken from one such part selected at random, provided the same specification and heat of material and the same process of production, including heat treatment, were used for all of the lot. When the part is too small to provide the three specimens of at least minimum size indicated in paragraph 3.11.7.2, then impact test do not need to be performed (see paragraph 3.11.7.2.c).

3.11.7.4 Certification of Compliance with Impact Test Requirements

a) Certified reports of impact tests by the materials manufacturer will be acceptable evidence that the material meets the requirements of this paragraph, provided:

    1) The specimens taken are representative of the material delivered (see paragraph 3.11.7.3.a) and the material is not subjected to heat treatment during or following fabrication that will materially reduce its impact properties; or

    2) The materials from which the specimens are removed are heat treated separately such that they are representative of the material in the finished vessel.

b) The Manufacturer of the vessel may have impact tests made to prove the suitability of a material which the materials manufacturer has not impact tested provided the number of tests and the method of taking the test specimens shall be as specified for the materials manufacturer.

3.11.7.5 Impact Test Temperature Criteria

For all Charpy impact tests, the following test temperature criteria shall be observed.

a) Materials of Thickness Equal to or Greater Than 10 mm (0.394 in.) – Where the largest obtainable Charpy V-notch specimen has a width along the notch of at least 8 mm (0.315 in.), the Charpy test of such a specimen shall be conducted at a temperature not warmer than the MDMT. Where the largest possible test specimen has a width along the notch less than 8 mm (0.315 in.), the test shall be conducted at a temperature colder than the MDMT by the amount shown in Table 3.11 for the specimen width. Note that this requirement does not apply when the option of paragraph 3.11.7.2.b is used.

b) Materials With Thickness Less Than 10 mm (0.394 in.) – Where the largest obtainable Charpy V-notch specimen has a width along the notch of at least 80% of the material thickness, the Charpy test of such a specimen shall be conducted at a temperature not warmer than the MDMT. Where the largest possible test specimen has a width along the notch of less than 80% of the material thickness, the test for carbon steel and low alloy materials having a specified minimum tensile strength of less than 655 MPa (95 ksi) shall be conducted at a temperature colder than the MDMT by an amount equal to the difference, see Table 3.11, between the temperature reduction corresponding to the actual material thickness and the temperature reduction corresponding to the Charpy specimen width actually tested. This requirement does not apply when the option of paragraph 3.11.7.2.b is used. For Table 3.A.2, carbon and low alloy materials having a specified minimum tensile strength greater than or equal to 655 MPa (95 ksi), for high alloy materials and quenched and tempered material with enhanced tensile properties, the test shall be conducted at a temperature not warmer than the MDMT.

3.11.7.6 Retests

a) Absorbed Energy Criteria – If the absorbed energy criteria are not met, retesting in accordance with the applicable procedures of SA-370 shall be permitted.

b) Lateral Expansion Criteria – retests shall be performed as follows:

    1) Retesting is permitted if the average value for three specimens equals or exceeds the value required.

        i) For materials of Table 3.A.1 (carbon and low alloy steels) having specified minimum tensile strengths of 655 MPa (95 ksi) or greater and for Table 3.A.2 (Q&T steels with enhanced strength properties) materials, if the measured value of lateral expansion for one specimen in a group of three is less than that required in Figure 3.6.

ii) For materials of Table 3.A.3 (high alloy steels) for MDMTs no colder than -196°C (-320°F), if the measured value of lateral expansion for one specimen in a group of three is less than 0.38 mm (0.015 in.), but not less than two-thirds of the value required.

iii) For materials of Table 3.A.3 (high alloy steels) for for MDMTs colder than -196°C (-320°F), if the value of lateral expansion for one specimen of a set is less than 0.53 mm (0.021 in.).

iv) For materials of Table 3.A.2 (Q&T steel with enhanced strength properties), if the measured value of lateral expansion for one specimen in a group of three is less than that required in Figure 3.6 but not less than two-thirds of the required value.

2) The retest shall consist of three additional specimens. For materials of Table 3.A.1 (carbon and low alloy steels) having specified minimum tensile strengths of 655 MPa (95 ksi) or greater and for Table 3.A.2 (Q&T steels with enhanced strength properties) materials, the retest value for each specimen must equal or exceed the value required in Figure 3.3 and Figure 3.4 for specimens not subject to PWHT and specimens subject to PWHT, respectively. For materials of Table 3.A.3 (high alloy steels), the retest value for each specimen must equal or exceed 0.38 mm (0.015 in.) for MDMTs no colder than -196°C (-320°F). For MDMTs colder than -196°C (-320°F), see paragraph 3.11.2.1.b.2 and 3.11.4.1.b.

3) In the case of materials with properties enhanced by heat treatment, the material may be reheat treated and retested if the required values are not obtained in the retest or if the values in the initial test are less than the values required for retest. After reheat treatment, a set of three specimens shall be made; for acceptance, the lateral expansion of each of the specimens must equal or exceed the value required in Figure 3.6.

c) When an erratic result is caused by a defective specimen or there is uncertainty in the test procedure, a retest will be allowed. When the option of paragraph 3.11.7.2.b is used for the initial test and the acceptance of 102 J (75 ft-lb) minimum is not attained, a retest using full-size (10 mm x 10 mm) specimens will be allowed.

3.11.8 Impact Testing Of Welding Procedures and Test Plates of Ferrous Materials

3.11.8.1 Impact Tests

a) For steel vessels of welded construction, the impact toughness of welds and heat affected zones of procedure qualification test plates and vessel test plates (production impact test plates) shall be determined as required in this paragraph.

b) All test plates shall be subjected to heat treatment, including cooling rates and aggregate time at temperature or temperatures as established by the manufacturer for use in actual manufacture. Heat treatment requirements of paragraph 6.4.2, paragraph 3.10.2, and paragraph 3.10.4 shall apply to test plates, except that the provisions of paragraph 3.10.3.2 are not applicable to test plates for welds joining P-No. 3, Groups 1 and 2 materials. For P-No. 1, Groups 1, 2, and 3 materials, impact testing of the welds and heat affected zones of the weld procedure qualification and production test plates need not be repeated if the impact properties were determined after PWHT of the test plates at the temperatures and times specified in Table 6.8 and the temperatures and times permitted in Table 6.16.

3.11.8.2 Location, Orientation, Temperature, and Values of Weld Impact Tests

a) All weld impact tests shall comply with the following requirements.

b) Each set of weld metal impact specimens shall be taken across the weld with the notch in the weld metal. Each specimen shall be oriented so that the notch is normal to the surface of the material, and one face of the specimen shall be within 1.5 mm (1/16 in.) of the surface of the material. When procedure tests are made on material over 38 mm (1-1/2 in.) in thickness, two sets of impact specimens shall be taken from the weld with one set located within 1.5 mm (1/16 in.) of the surface of one side of the material and one set taken as near as practical midway between the surface and the center of thickness of the opposite side as described above [see QW-200.4(a) of Section IX].

c) Each set of heat affected zone impact specimens shall be taken across the weld and of sufficient length to locate, after etching, the notch in the affected zone. The notch shall be cut approximately normal to the material surface in such a manner as to include as much heat affected zone material as possible in the resulting fracture.

d) For welds made by a solid-state welding process, such as for electric resistance welded (ERW) pipe, the

weld impact tests shall consist only of one set of three specimens taken across the weld with the notch at the weld centerline. Each specimen shall be oriented so that the notch is normal to the surface of the material and one face of the specimen shall be within 1.5 mm (1/16 in.) of the surface of the material.
e) The test temperature for welds and heat affected zones shall not be higher than for the base materials.
f) Impact values shall be at least as high as those required for the base materials (see paragraphs 3.11.2, 3.11.3, and 3.11.4, as applicable).

3.11.8.3 Impact Tests for Welding Procedures

a) Welding procedure impact tests shall be made on welds and heat affected zones when base materials are required to be impact tested, except as exempted by paragraph 3.11.4.4 and 3.11.2.10.
b) If impact tests are required for the deposited weld, but the base material is exempted from impact tests, welding procedure test plates shall be made. The test plate material shall be material of the same P-Number and Group Number used in the vessel. One set of impact specimens shall be taken with the notch approximately centered in the weld metal and perpendicular to the surface; the heat affected zone need not be impact tested.
c) When the welding procedure employed for production welding is used for fillet welds only, it shall be qualified by a groove weld qualification test. The qualification test plate or pipe material shall meet the requirements of paragraph 3.11.7 when impact testing is a requirement. This welding procedure test qualification is in addition to the requirements of Section IX, QW-202.2 for P-No. 11 materials.
d) The supplementary essential variables specified in Section IX, QW-250, for impact testing are required.
e) For test plates or pipe receiving a postweld heat treatment in which the lower critical temperature is exceeded, the maximum thickness qualified is the thickness of the test plate or pipe.
f) For materials of Table 3.A.1 (carbon steel and low alloy steel), the test plate material shall satisfy all of the following requirements relative to the material to be used in production:
   1) Be of the same P-Number and Group Number;
   2) Be in the same heat treated condition;
   3) Meet the minimum toughness requirements paragraphs 3.11.2, 3.11.3, and 3.11.4, as applicable for the thickest material of the range of base material qualified by the procedure.

3.11.8.4 Impact Tests of Vessel Test Plates

a) When the base material is required to be impact tested, impact tests of welds and heat affected zones shall be made for Category A and B joints in accordance with paragraph 3.11.8.2 for each qualified welding procedure used on each vessel. The test plate shall be from one of the heats of steel used for the vessel or group of vessels and shall be welded as an extension to the end of a production Category A joint where practicable, or welded as close to the start of production welding as practicable, utilizing equipment, welding materials, and procedures which are to be used on the production joint.
b) For Category B joints that are welded using a different welding procedure than used on Category A joints, a test plate shall be welded under the production welding conditions used for the vessel, using the same type of equipment and at the same location and using the same procedures as used for the joint, and it shall be welded concurrently with the production welds or as close to the start of production welding as practicable.
c) Number Of Vessel Impact Test Plates Required
   1) For each vessel, one test plate shall be made for each welding procedure used for joints of Categories A and B, unless the vessel is one of several as defined in paragraph 3.11.8.4.c.2. In addition, for Category A and B joints, the following requirements shall apply:
      i) If automatic or semiautomatic welding is performed, a test plate shall be made in each position employed in the vessel welding.
      ii) If manual welding is also employed, a test plate shall be made in the flat position only, except if welding is to be performed in other positions a test plate need be made in the vertical position only (where the major portions of the layers of welds are deposited in the vertical upward direction). The vertically welded test plate will qualify the manual welding in all positions.
      iii) The vessel test plate shall qualify the impact requirements for vessel materials thickness in accordance with Section XI, paragraphs QW-45l.1 and QW-451 (including Notes), except

that, if the thickness is less than 16 mm (5/8 in.), the thickness of the test material is the minimum thickness qualified.

2) For several vessels or parts of vessels, welded within any 3 month period at one location, the plate thickness of which does not vary by more than 6 mm (1/4 in.) or 25%, whichever is greater, and of the same specification and grade of material, a test plate shall be made for each 122 m (400 ft) of joints welded by the same procedure.

d) If the vessel test plate fails to meet the impact requirements, the welds represented by the test plate shall be unacceptable. Reheat treatment and retesting, or retesting only, are permitted.

3.12 Allowable Design Stresses

The design stresses for materials permitted by this Division are given in Annex 3.A.

3.13 Strength Parameters

The strength parameters for materials permitted by this Division are given in Annex 3.D

3.14 Physical Properties

The following physical properties for all permissible materials of construction are given in the tables referenced in Annex 3.E.

a) Young's Modulus
b) Thermal Expansion Coefficient
c) Thermal Conductivity
d) Thermal Diffusivity

3.15 Design Fatigue Curves

Design fatigue curves for non-welded and for welded construction are provided in Annex 3.F. As an alternative, the adequacy of a part to withstand cyclic loading may be demonstrated by means of fatigue test following the requirements of Annex 5.F. However, the fatigue test shall not be used as justification for exceeding the allowable values of primary or primary plus secondary stresses.

3.16 Nomenclature

a reference flaw depth.

$2c$ reference flaw length.

E joint efficiency (see Part 7) used in the calculation of t_r. For castings, the quality factor or joint efficiency E, whichever governs design, shall be used.

E^* E^* equal to E except that E^* shall not be less than 0.80, or $E^* = \max\left[E,\ 0.80\right]$.

CA corrosion allowance

$MDMT$ Minimum Design Metal Temperature.

P_a applied pressure for the condition under consideration.

P_{rating} maximum allowable working pressure based on the design rules in this Division of ASME/ANSI pressure-temperature ratings.

R_{ts} stress ratio defined as the stress for the operating condition under consideration divided by the stress at the design minimum temperature. The stress ratio may also be defined in terms of required and actual thicknesses, and for components with pressure temperature ratings, the stress ratio is computed as the applied pressure for the condition under consideration divided by the pressure rating at the $MDMT$.

S allowable stress from Annex 3.A

S_y specified minimum yield strength.

S^* applied general primary stress.

t reference flaw plate thickness.

t_g governing thickness.

t_n nominal uncorroded thickness. For welded pipe where a mill undertolerance is allowed by the material specification, the thickness after mill undertolerance has been deducted shall be taken as the nominal thickness. Likewise, for formed heads, the minimum specified thickness after forming shall be used as the nominal thickness.

t_r required thickness of the part under consideration in the corroded condition for all applicable loadings

T_R reduction in $MDMT$ based on available excess thickness.

3.17 Definitions

The definitions for the terminology used in this Part are contained in Annex 1.B.

3.18 Tables

Table 3.1 – Material Specifications

Nominal Composition	Type/Grade	Specification	Product Form
2¼Cr-1Mo	Grade 22, Cl. 3	SA-508	Forgings
	Grade 22, Cl. 3	SA-541	Forgings
	Type B, Cl. 4	SA-542	Plates
2¼Cr-1Mo-¼V	Grade F22V	SA-182	Forgings
	Grade F22V	SA-336	Forgings
	Grade 22V	SA-541	Forgings
	Type D, Cl. 4a	SA-542	Plates
	Grade 22V	SA-832	Plates
3Cr-1Mo-¼V-Ti-B	Grade F3V	SA-182	Forgings
	Grade F3V	SA-336	Forgings
	Grade 3V	SA-508	Forgings
	Grade 3V	SA-541	Forgings
	Type C, Cl. 4a	SA-542	Plates
	Grade 21 V	SA-832	Plates

Table 3.2 – Composition Requirements For 2.25Cr-1Mo-0.25V Weld Metal

Welding Process	C	Mn	Si	Cr	Mo	P	S	V	Cb
SAW	0.05-0.15	0.50-1.30	0.05-0.35	2.00-2.60	0.90-1.20	0.015 max	0.015 max	0.20-0.40	0.010-0.040
SMAW	0.05-0.15	0.50-1.30	0.20-0.50	2.00-2.60	0.90-1.20	0.015 max	0.015 max	0.20-0.40	0.010-0.040
GTAW	0.05-0.15	0,30-1.10	0.05-0.35	2.00-2.60	0.90-1.20	0.015 max	0.015 max	0.20-0.40	0.010-0.040
GMAW	0.05-0.15	0.30-1.10	0.20-0.50	2.00-2.60	0.90-1.20	0.015 max	0.015 max	0.20-0.40	0.010-0.040

Table 3.3 – Toughness Requirements for 2.25Cr-1Mo Materials

Number of Specimens	Impact Energy, Joules (ft-lb)
Average of 3	54 (40)
Only one in the set	48 (35)
Note: Full size Charpy V-notch, transverse, tested at the MDMT.	

Table 3.4 – Low Alloy Bolting Materials For Use With Flanges Designed To Part 4, Paragraph 4.16

Material Specification	Material Type/Grade	Diameter mm (in)	MDMT Without Impact Testing °C (°F)
Low Alloy Bolting			
SA-193	B5	Up to 102 (4), inclusive	-29 (-20)
	B7	64 (2-1/2) and under	-48 (-55)
		Over 64 to 102 (2-1/2 to 4), inclusive	-40 (-40)
		Over 102 to 178 (4 to 7), inclusive	-40 (-40)
	B7M	64 (2-1/2) and under	-48 (-55)
	B16	64 (2-1/2) and under	-29 (-20)
		Over 64 to 102 (2-1/2 to 4), inclusive	-29 (-20)
		Over 102 to 178 (4 to 7), inclusive	-29 (-20)
SA-320	L7	64 (2-1/2) and under	See paragraph 3.11.2.4.b
	L7 A	Up to 64 (2-1/2), inclusive	See paragraph 3.11.2.4.b
	L7M	64 (2-1/2) and under	See paragraph 3.11.2.4.b
	L43	25 (1) and under	See paragraph 3.11.2.4.b
SA-325	1	13 to 38 (1/2 to 1-1/2), inclusive	-29 (-20)
SA-354	BC	Up to 102 (4),	-18 (0)
	BD	Up to 102 (4), inclusive	-7 (+20)
SA-437	B4B, B4C	All diameters	See paragraph 3.11.2.4.b
SA-449	---	Up to 76 (3), inclusive	-29 (-20)
SA-508	5 Cl.2	All diameters	See paragraph 3.11.2.4.b
SA-540	B21	All diameters	Impact test is required
	B23 Cl. 1 & 2	All diameters	Impact test is required
	B23 Cl. 3 & 4	Up to 152 (6), inclusive	See paragraph 3.11.2.4.b
		Over 152 to 241 (6 to 9-1/2), inclusive	Impact test is required
	B23 Cl. 5	Up to 203 (8), inclusive	See paragraph 3.11.2.4.b
		Over 203 to 241 (8 to 9-1/2), inclusive	Impact test is required
	B24 Cl. 1	Up to 152 (6), inclusive	See paragraph 3.11.2.4.b
		Over 152 to 203 (6 to 8), inclusive	Impact test is required
	B24 Cl. 2	Up to 178 (7), inclusive	See paragraph 3.11.2.4.b
		Over 178 to 241 (7 to 9-1/2), inclusive	Impact test is required
	B24 Cl. 3 & 4	Up to 203 (8), inclusive	See paragraph 3.11.2.4.b
		Over 203 to 241 (8 to 9-1/2), inclusive	Impact test is required
	B24 Cl. 5	Up to 241 (9-1/2), inclusive	See paragraph 3.11.2.4.b
	B24V Cl. 3	All diameters	See paragraph 3.11.2.4.b
Low Alloy Steel Nuts			
SA-194	2, 2H, 2HM, 3, 4, 7, 7M, 16	All diameters	-48 (-55)
SA-540	B21, B23, B24, B24V	All diameters	-48 (-55)

Table 3.5 – High Alloy Bolting Materials For Use With Flanges Designed To Part 4, Paragraph 4.16

Material Specification	Material Type/Grade	Diameter mm (in)	MDMT Without Impact Testing °C (°F)
SA-193	B6	102 (4) and under	-29 (-20)
	B8 Cl. 1	All diameters	-254 (-425)
	B8 Cl. 2	Up to 38 (1-1/2), inclusive	Impact test is required
	B8C Cl. 1	All diameters	-254 (-425)
	B8C Cl. 2	19 to 38 (0.75 to 1-1/2), inclusive	Impact test is required
SA-193	B8M Cl. 1	All diameters	-254 (-425)
	B8M2	51 to 64 (2 to 2-1/2), inclusive	Impact test is required
	B8MNA Cl. 1A	All diameters	-196 (-320)
	B8NA Cl. 1A	All diameters	-196 (-320)
	B8P Cl. 1	All diameters	Impact test is required
	B8P Cl. 2	Up to 38 (1-1/2), inclusive	Impact test is required
	B8S, 88SA	All diameters	Impact test is required
	B8T Cl. 1	All diameters	-254 (-425)
	B8T Cl, 2	19 to 25 (3/4 to 1), inclusive	Impact test is required
SA-320	B8 Cl. 1	All diameters	See paragraph 3.11.2.4.b
	B8 Cl. 2	Up to 25 (1), inclusive	See paragraph 3.11.2.4.b
	B8A Cl. 1A	All diameters	See paragraph 3.11.2.4.b
	B8C Cl. 1 & 1A	All diameters	See paragraph 3.11.2.4.b
	B8C Cl. 2	Up to 25 (1), inclusive	See paragraph 3.11.2.4.b
	B8CA Cl. 1A	All diameters	See paragraph 3.11.2.4.b
	B8F Cl. 1	All diameters	See paragraph 3.11.2.4.b
	B8FA Cl.1A	All diameters	See paragraph 3.11.2.4.b
	B8M Cl. 1	All diameters	See paragraph 3.11.2.4.b
	B8M Cl. 2	Up to 38 (1-1/2), inclusive	See paragraph 3.11.2.4.b
	B8MA Cl. 1A	All diameters	See paragraph 3.11.2.4.b
	B8T Cl. 1	All diameters	See paragraph 3.11.2.4.b
	B8T Cl. 2	Up to 38 (1-1/2), inclusive	See paragraph 3.11.2.4.b
	B8TA Cl. 1A	All diameters	See paragraph 3.11.2.4.b
SA-453	651 Cl. A & B, 660 Cl. A & B	All diameters	Impact test is required
SA-479	XM-19	Up to 8 (203), inclusive	Impact test is required
SA-564	630	Up to 8 (203), inclusive.	Impact test is required
SA-705	630	Up to 8 (203), inclusive.	Impact test is required

Table 3.6 – Aluminum Alloy, Copper, and Copper Alloy Bolting Materials For Use With Flanges Designed To Part 4, Paragraph 4.16

Material Specification	UNS
SB-98	C65100, C65500, C66100
SB-150	C61400, C62300, C63000, C64200
SB-187	C10200, C11000
SB-211	A92014, A92024, A96061
Note: The MDMT for all bolting material listed in this Table is -196°C (-320°F).	

Table 3.7 – Nickel and Nickel Alloy Bolting Materials For Use With Flanges Designed To Part 4, Paragraph 4.16

Material Specification	UNS
SB-160	N02200, N02201
SB-164	N04400 N04405
SB-166	N06600
SB-335	N10001, N10665
SB-408	N08800, N08810
SB-425	N08825
SB-446	N06625
SB-572	N06002, R30556
SB-573	N10003
SB-574,	N06022, N06455, N10276
SB-581	N06007, N06030, N06975
SB-621	N08320
SB-637	N07718, N07750
Note: The MDMT for all bolting material listed in this Table is -196°C (-320°F).	

Table 3.8 – Bolting Materials For Use With Flanges Designed To Part 5

Material Specification	Material Grade
SA-193	B5, B6, B7, B7M, B8, B8C, B8M, B8MNA, B8NA, B8R, B8RA, B8S, B8SA, B8T, B16
SA-320	L43
SA-437	B4B, B4C
SA-453	651, 660
SA-540	B21, B22, 823, B24, B24V
SA-564	630
SA-705	630
SB-164	N04400, N04405
SB-637	N07718, N07750
Note: See paragraph 3.11.6.2 for impact testing requirements	

Table 3.9 – Maximum Severity Levels For Castings With A Thickness Of Less Than 50 mm (2 in.)

Imperfection Category	Thickness < 25 mm (1 in.)	Thickness 25 mm < 50 mm (1 in. < 2 in.)
A – Gas porosity	1	2
B – Sand and slag	2	3
C – Shrinkage (four types)	1	3
D – Cracks	0	0
E – Hot tears	0	0
F - Inserts	0	0
G – Mottling	0	0

Table 3.10 – Maximum Severity Levels For Castings With A Thickness Of 50-305 mm (2-12 in.)

Imperfection Category	Thickness 50 mm to 115 mm (2 in. to 4-1/2 in.)	Thickness >115 mm to 305 mm (> 4-1/2 in. to 12 in.)
A - Gas porosity	2	2
B – Sand and slag inclusions	2	2
C – Shrinkage – Type 1	1	2
C – Shrinkage – Type 2	2	2
C – Shrinkage – Type 3	3	2
D – Cracks	0	0
E – Hot tears	0	0

Table 3.11 – Charpy Impact Test Temperature Reduction Below The Minimum Design Metal Temperature (1)

Actual Material Thickness (see Paragraph 3.11.7.5.b) or Charpy Impact Specimen Width Along the Notch		Temperature Reduction	
mm	in.	°C	°F
10 (full-size standard bar)	0.394	0	0
9	0.354	0	0
8	0.315	0	0
7.5 (3/4 size bar)	0.295	3	5
7	0.276	4	8
6.65 (2/3 size bar)	0.262	6	10
6	0.236	8	15
5 (1/2 size bar)	0.197	11	20
4	0.158	17	30
3.33 (1/3 size bar)	0.131	19	35
3	0.118	22	40
2.5 (1/4 size bar)	0.099	28	50

Notes:

1) For carbon and low alloy materials having a specified minimum tensile strength of less than 655 MPa (95 ksi) when the subsize Charpy impact width is less than 80% of the material thickness
2) Straight line interpolation for intermediate values is permitted.

Table 3.12 – Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels As a Function of the Minimum Specified Yield Strength – Parts Not Subject to PWHT (see Figure 3.3 and 3.3M)

Thickness (mm)	CVN (J)					Thickness (in)	CVN (ft-lbs)				
	Specified Minimum Yield Strength (MPa)						Specified Minimum Yield Strength (ksi)				
	205	260	345	450	550		30	38	50	65	80
6	27	27	27	27	27	**0.25**	20	20	20	20.	20
10	27	27	27	27	31	**0.375**	20	20.	20	20	23
13	27	27	27	27	36	**0.5**	20	20	20	20	27
16	27	27	27	29	43	**0.625**	20	20	20	21	32
19	27	27	27	34	51	**0.75**	20	20	20	25	37
25	27	27	27	45	62	**1**	20	20	20	33	46
32	27	27	34	53	72	**1.25**	20	20	25	39	53
38	27	27	40	61	82	**1.5**	20	20	30	45	60

Note: The Charpy V-notch values given in this table represent a smooth curve in Figure. 3.3 and 3.3M.

Table 3.13 – Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels As a Function of the Minimum Specified Yield Strength – Parts Subject to PWHT (see Figure 3.4 and 3.4M)

Thickness (mm)	CVN (J) Specified Minimum Specified Yield Strength (MPa)					Thickness (in)	CVN (ft-lbs) Specified Minimum Specified Yield Strength (ksi)				
	205	260	345	450	550		30	38	50	65	80
6	27	27	27	27	27	0.25	20	20	20	20	20
10	27	27	27	27	27	0.375	20	20	20	20	20
13	27	27	27	27	27	0.5	20	20	20	20	20
16	27	27	27	27	27	0.625	20	20	20	20	20
19	27	27	27	27	27	0.75	20	20	20	20	20
25	27	27	27	27	27	1	20	20	20	20	20
32	27	27	27	27	34	1.25	20	20	20	20	25
38	27	27	27	27	40	1.5	20	20	20	20	30
44	27	27	27	31	47	1.75	20	20	20	23	35
51	27	27	27	35	52	2	20	20	20	26	38
57	27	27	27	40	56	2.25	20	20	20	29	41
64	27	27	27	43	60	2.5	20	20	20	32	44
70	27	27	29	46	64	2.75	20	20	21	34	47
76	27	27	31	49	68	3	20	20	23	36	50
83	27	27	33	52	71	3.25	20	20	25	38	52
89	27	27	35	54	74	3.5	20	20	26	40	54
95	27	27	37	56	76	3.75	20	20	27	42	56
102	27	27	38	58	78	4	20	20	28	43	58.
108	27	27	39	59	80	4.25	20	20	29	44	59
114	27	27	40	60	81	4.5	20	20	29	45	60
121	27	27	40	61	82	4.75	20	20	30	45	60
127	27	27	41	61	82	5	20	20	30	45	61
133	27	27	41	61	82	5.25	20	20	30	45	61
140	27	27	41	61	82	5.5	20	20	30	45.	61
146	27	27	41	61	82	5.75	20	20	30	45	61
152	27	27	41	61	82	6	20	20	30	45	61
159	27	27	41	61	82	6.25	20	20	30	45	61
165	27	27	41	61	82	6.5	20	20	30	45	61
171	27	27	41	61	82	6.75	20	20	30	45	61
178	27	27	41	61	82	7	20	20	30	45	61

Note: The Charpy V-notch values given in this table represent a smooth curve in Figure 3.4 and 3.4M.

Table 3.14 – Impact Test Exemption Curves – Parts Not Subject to PWHT (see Figure 3.7 and 3.7M)

Thickness (mm)	Exemption Curve (deg. C)				Thickness (in)	Exemption Curve (deg. F)			
	A	B	C	D		A	B	C	D
0	20.5	-0.6	-21.7	-36.1	0	68.9	30.9	-7.1	-33.1
10	20.5	-0.6	-21.7	-36.1	0.394	68.9	30.9	-7.1	-33.1
13	22.9	1.8	-19.3	-33.7	0.5	73.3	35.3	-2.7	-28.7
16	26.3	5.1	-16.0	-30.4	0.625	79.3	41.3	3.3	-22.7
19	29.6	8.5	-12.6	-27.1	0.75	85.3	47.3	9.3	-16.7
25	35.2	14.1	-7.0	-21.4	1	95.4	57.4	19.4	-6.6
32	39.7	18.6	-2.6	-17.0	1.25	103.4	65.4	27.4	1.4
38	43.4	22.3	1.2	-13.2	1.5	110.2	72.2	34.2	8.2
Note: The Charpy V-notch values given in this table represent a smooth curve in Figure 3.7 and 3.7M.									

Table 3.15 – Impact Test Exemption Curves – Parts Subject to PWHT and Non-welded Parts (see Figure 3.8 and 3.8M)

Thickness (mm)	Exemption Curve (deg. C)				Thickness (in)	Exemption Curve (deg. F)			
	A	B	C	D		A	B	C	D
0	0.6	-20.5	-41.6	-48.3	0	33.2	-4.8	-42.8	-55.0
10	0.6	-20.5	-41.6	-48.3	0.394	33.2	-4.8	-42.8	-55.0
13	3.8	-17.3	-38.4	-48.3	0.5	38.9	0.9	-37.1	-55.0
16	7.9	-13.2	-34.3	-48.3	0.625	46.2	8.2	-29.8	-55.0
19	11.7	-9.4	-30.5	-45.0	0.75	53.0	15.0	-23.0	-49.0
25	17.5	-3.6	-24.7	-39.2	1	63.5	25.5	-12.5	-38.5
32	21.7	0.5	-20.6	-35.0	1.25	71.0	33.0	-5.0	-31.0
38	24.9	3.8	-17.3	-31.8	1.5	76.8	38.8	0.8	-25.2
44	27.7	6.6	-14.6	-29.0	1.75	81.8	43.8	5.8	-20.2
51	30.1	9.0	-12.1	-26.5	2	86.2	48.2	10.2	-15.8
57	32.4	11.3	-9.9	-24.3	2.25	90.3	52.3	14.3	-11.7
64	34.4	13.3	-7.8	-22.3	2.5	93.9	55.9	17.9	-8.1
70	36.2	15.1	-6.0	-20.5	2.75	97.2	59.2	21.2	-4.8
76	37.8	16.7	-4.4	-18.9	3	100.0	62.0	24.0	-2.0
83	39.2	18.1	-3.0	-17.5	3.25	102.6	64.6	26.6	0.6
89	40.4	19.3	-1.8	-16.3	3.5	104.7	66.7	28.7	2.7
95	41.4	20.3	-0.8	-15.3	3.75	106.5	68.5	30.5	4.5
102	42.2	21.1	-0.1	-14.5	4	107.9	69.9	31.9	5.9
Note: The Charpy V-notch values given in this table represent a smooth curve in Figure 3.8 and 3.8M.									

Table 3.16 – Reduction in the MDMT (T_R) without Impact Testing – Parts Not Subject to PWHT (see Figure 3.12 and 3.12M)

Stress or Thickness Ratio	T_R (°C)		T_R (°F)	
	Specified Minimum Yield Strength (MPa)		Specified Minimum Yield Strength (ksi)	
	$\leq 345\ MPa$	$> 345\ MPa,\ \leq 450\ MPa$	$\leq 50\ ksi$	$> 50\ ksi,\ \leq 65\ ksi$
1.000	0.0	0.0	0.0	0.0
0.940	2.7	2.5	4.9	4.5
0.884	5.2	4.7	9.3	8.4
0.831	7.3	6.6	13.2	11.9
0.781	9.3	8.4	16.7	15.1
0.734	11.1	10.0	20.0	18.1
0.690	12.8	11.5	23.0	20.8
0.648	14.3	13.0	25.8	23.3
0.610	15.8	14.3	28.5	25.7
0.573	17.2	15.5	31.0	27.9
0.539	18.5	16.7	33.3	30.0
0.506	19.7	17.7	35.5	31.9
0.476	20.9	18.8	37.6	33.8
0.447	22.0	19.7	39.6	35.5
0.421	23.1	20.6	41.5	37.1
0.395	24.0	21.5	43.3	38.7
0.372	25.0	22.3	45.0	40.1
0.349	25.9	23.1	46.6	41.5
0.328	26.7	23.8	48.1	42.8
0.309	27.5	24.5	49.6	44.0
0.290	28.3	25.1	50.9	45.2
0.273	29.0	25.7	52.2	46.3
0.256	29.7	26.3	53.5	47.3
0.241	30.4	26.8	54.6	48.3

Note: The temperature reduction values given in this table represent a smooth curve in Figure 3.12 and 3.12M.

Table 3.17 – Reduction in the MDMT (T_R) without Impact Testing - Parts Subject to PWHT and Non-welded Parts (see Figure 3.13 and 3.13M)

Stress or Thickness Ratio	T_R (°C)		T_R (°F)	
	Specified Minimum Yield Strength (MPa)		Specified Minimum Yield Strength (ksi)	
	$\leq 345\ MPa$	$> 345\ MPa,$ $\leq 450\ MPa$	$\leq 50\ ksi$	$> 50\ ksi,$ $\leq 65\ ksi$
1.000	0.0	0.0	0.0	0.0
0.940	3.0	2.6	5.4	4.6
0.884	5.9	5.0	10.6	8.9
0.831	8.7	7.3	15.6	13.1
0.781	11.5	9.5	20.7	17.2
0.734	14.3	11.7	25.8	21.1
0.690	17.3	13.9	31.1	25.0
0.648	20.3	16.1	36.5	29.0
0.610	23.5	18.3	42.2	32.9
0.573	26.9	20.5	48.4	36.8
0.539	30.6	22.7	55.0	40.9
0.506	34.7	25.0	62.5	45.0
0.476	39.5	27.3	71.1	49.2
0.447	45.3	29.8	81.6	53.6
0.421	52.9	32.3	95.2	58.1
0.395	---	35.0	---	62.9
0.372	---	37.8	---	68.1
0.349	---	40.9	---	73.6
0.328	---	44.3	---	79.7
0.309	---	48.0	---	86.4
0.290	---	52.3	---	94.2
0.273	---	---	---	
0.256	---	---	---	---
0.241	---	---	---	---

Note: The temperature reduction values given in this table represent a smooth curve in Figure 3.13 and 3.13M.

3.19 Figures

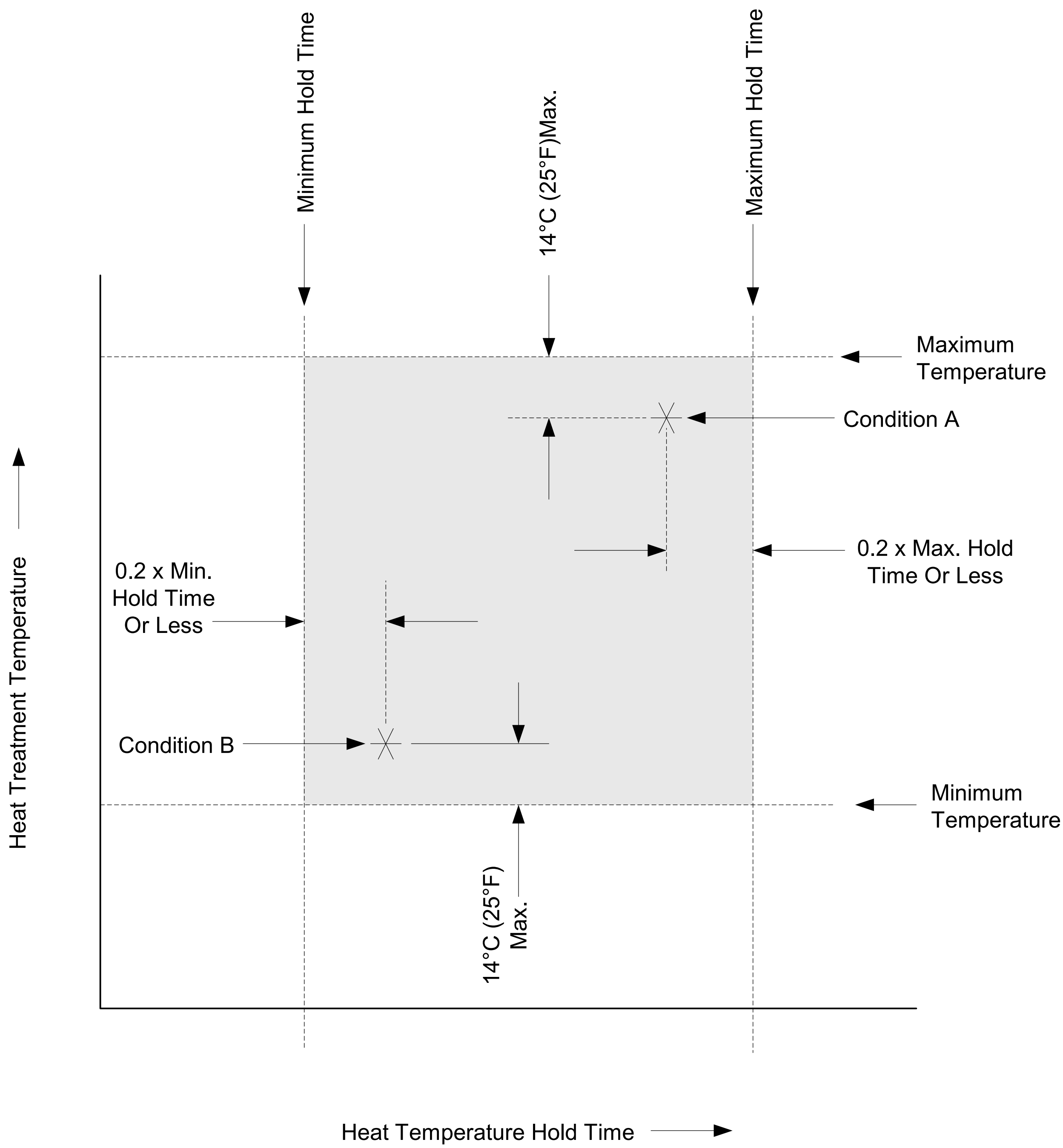
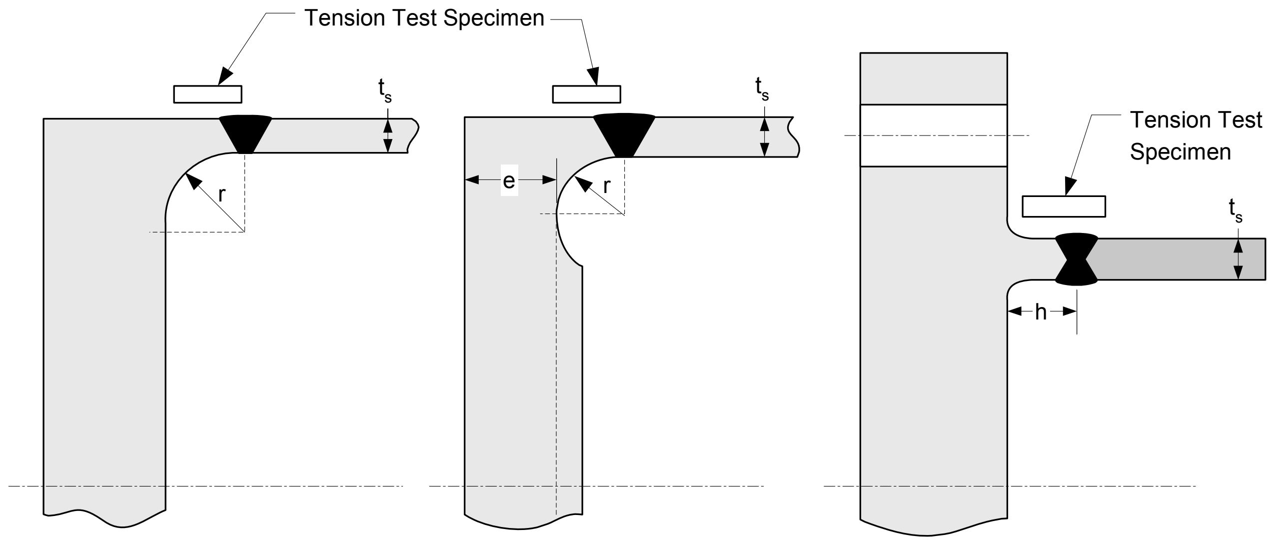


Figure 3.1 – Cr-Mo Heat Treatment Criteria



Note: These details are not permissible if machined from plate unless the requirements of paragraph 3.9 are satisfied.

Figure 3.2 – Typical Locations for Tensile Specimens

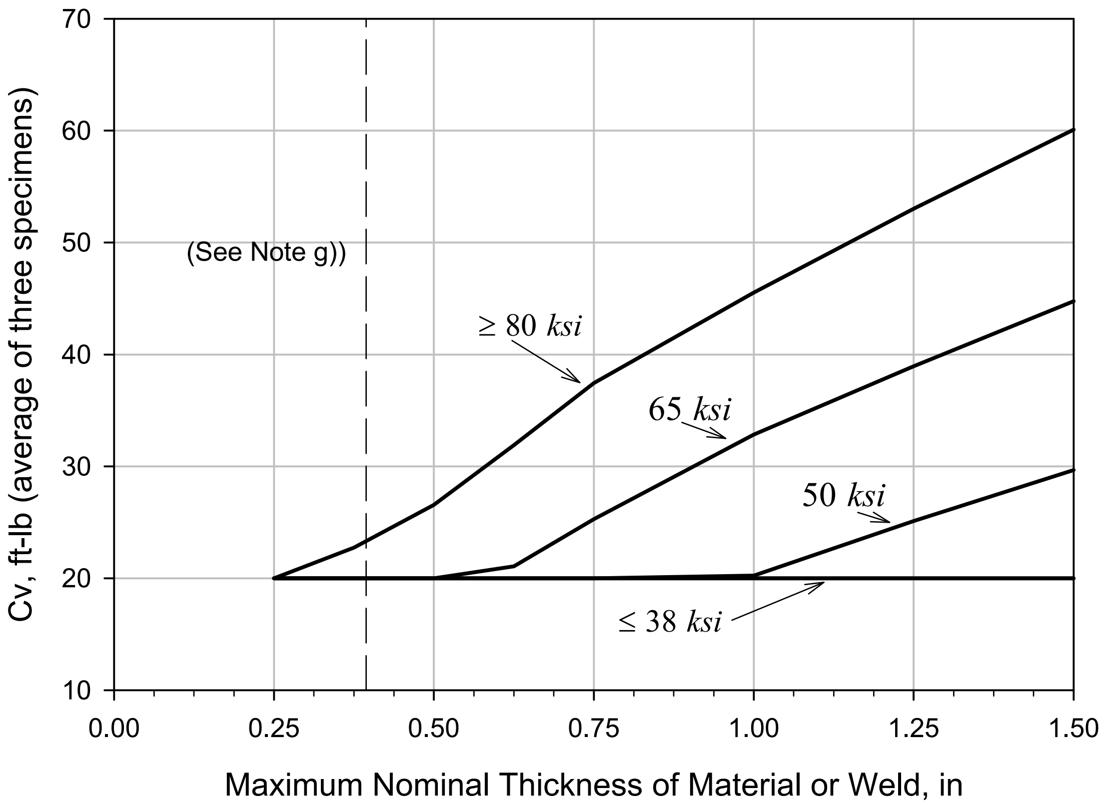


Figure 3.3 – Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels As a Function of the Minimum Specified Yield Strength – Parts Not Subject to PWHT

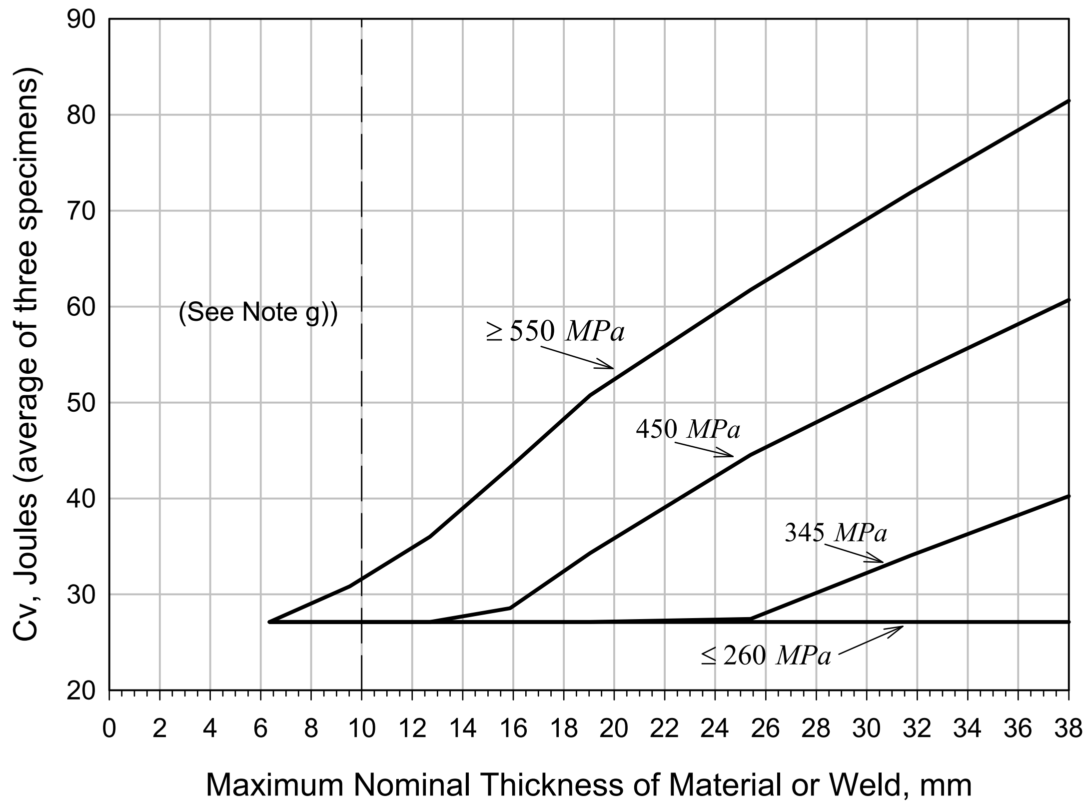


Figure 3.3M – Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels As a Function of the Minimum Specified Yield Strength – Parts Not Subject to PWHT

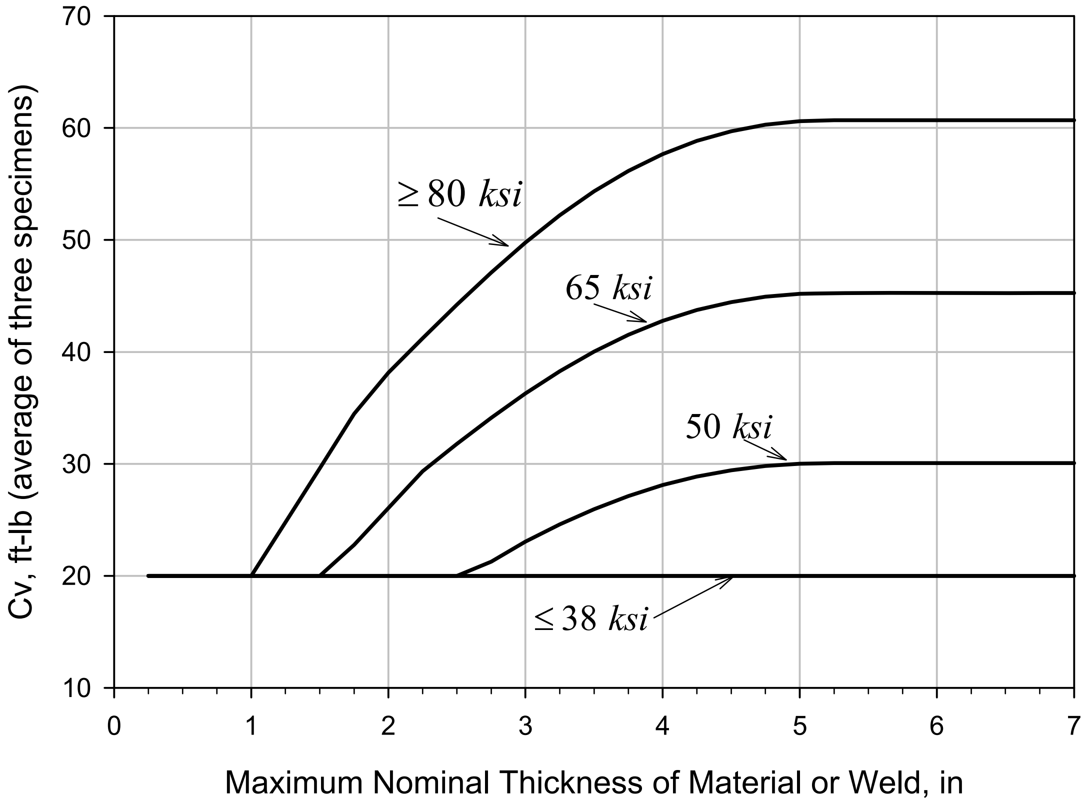


Figure 3.4 – Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels As a Function of the Minimum Specified Yield Strength – Parts Subject to PWHT

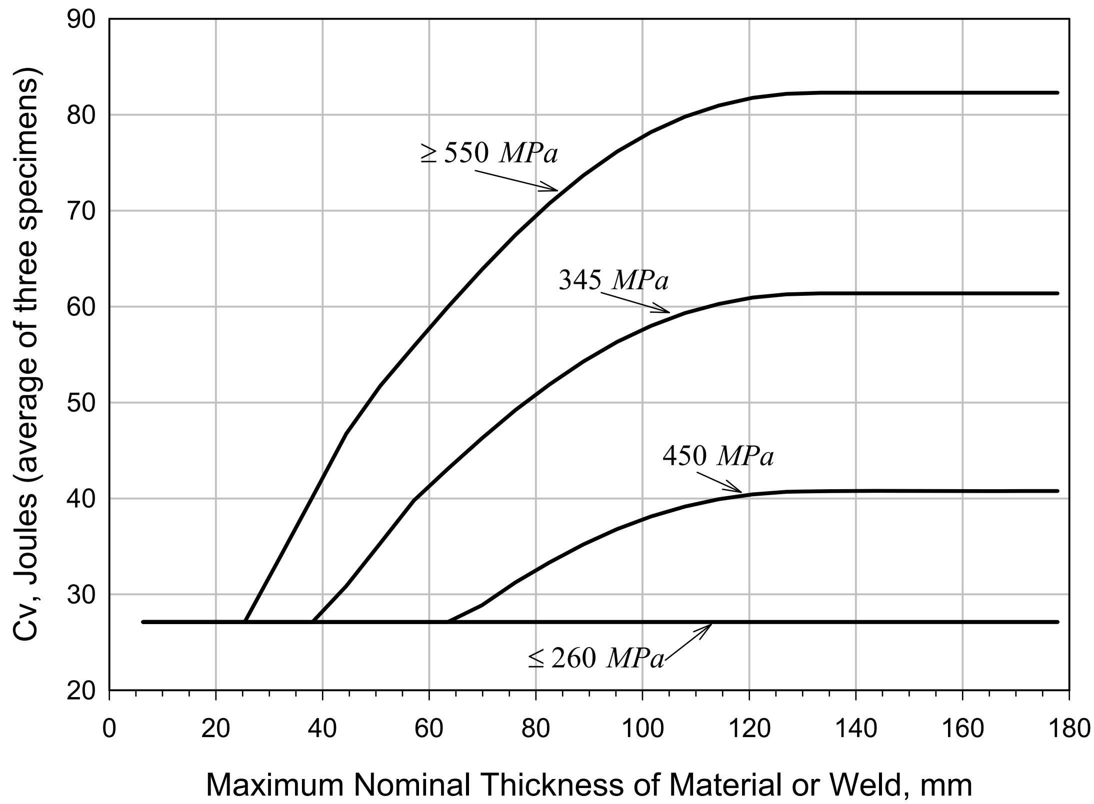


Figure 3.4M – Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels As a Function of the Minimum Specified Yield Strength – Parts Subject to PWHT

Notes for Figures 3.3, 3.3M, 3.4, and 3.4M

a) Interpolation between yield strength values is permitted.

b) The minimum impact energy for one specimen shall not be less than two-thirds of the average impact energy required for three specimens.

c) Materials produced and impact tested in accordance with SA-320, SA-333, SA-334, SA-350, SA-352, SA-420, SA-437, SA-508 Grade 5 Class 2, SA-540 (except for materials produced under Table 2, Note 4 in the specification), SA-723, and SA-765 do not have to satisfy these energy values. Materials produced to these specifications are acceptable for use at a minimum design metal temperature not colder than the test temperature when the energy values required by the applicable specification are satisfied.

d) If the material specified minimum tensile strength is greater than or equal to 655 MPa (95 ksi), then the material toughness requirements shall be in accordance with paragraph 3.11.2.1.b.2.

e) Data of Figures 3.3 and 3.3M are shown in Table 3.12.

f) Data of Figures 3.4 and 3.4M are shown in Table 3.13.

g) See paragraph 3.11.2.1.b.1 for Charpy V-notch specimen thicknesses less than 10 mm (0.394 in.)

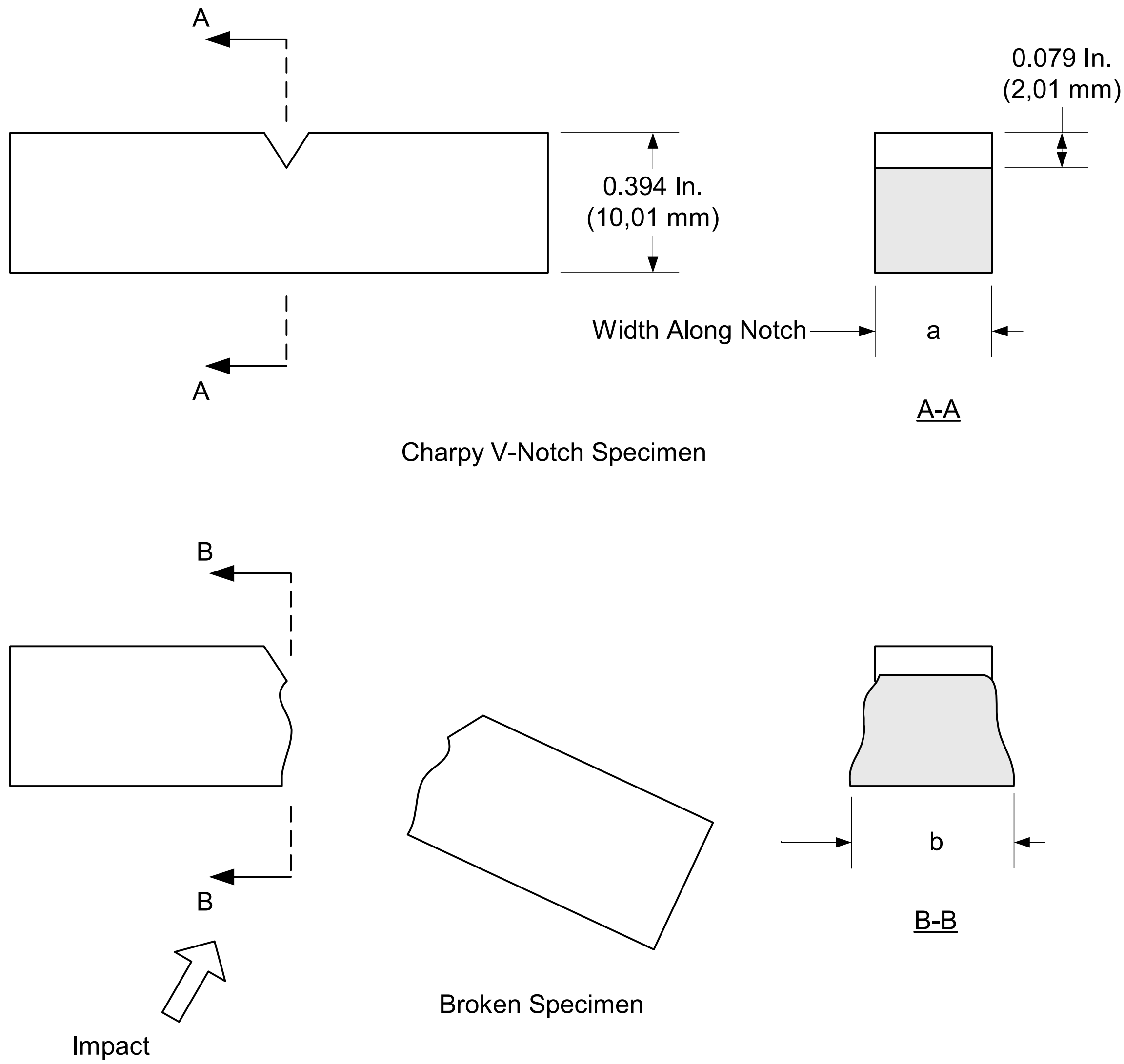


Figure 3.5 – Illustration of Lateral Expansion in a Broken Charpy V-Notch Specimen

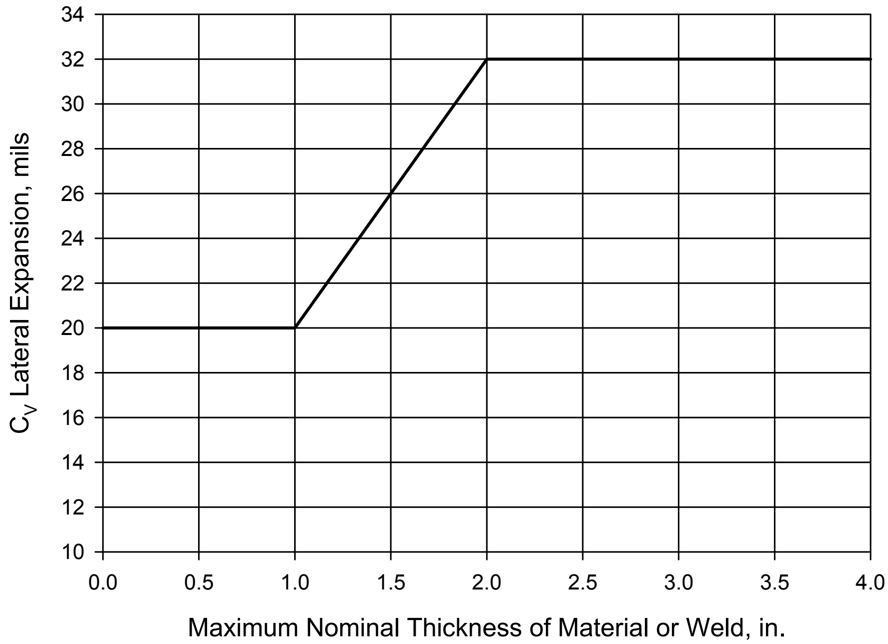


Figure 3.6 – Lateral Expansion Requirements

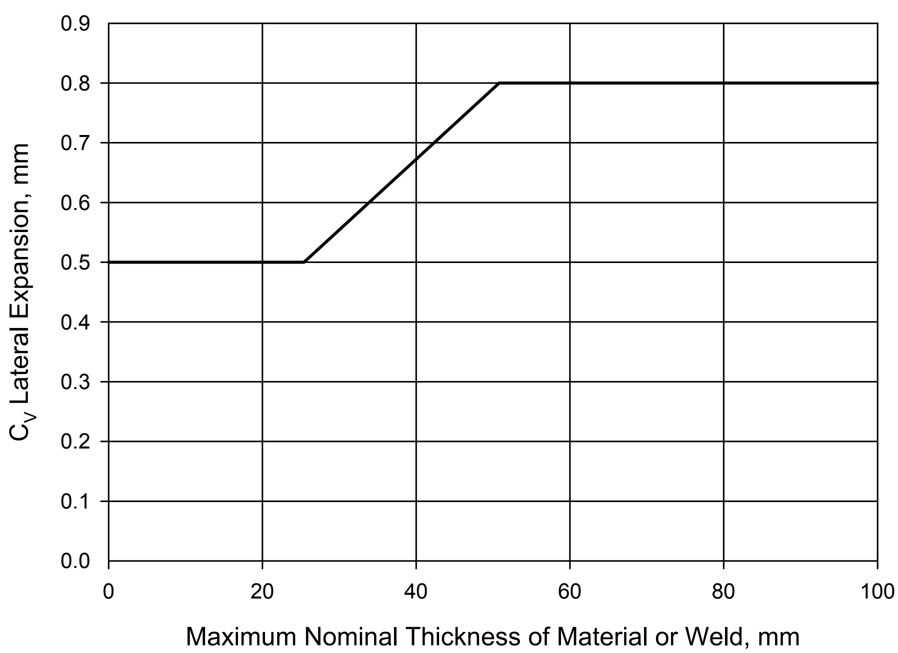


Figure 3.6M – Lateral Expansion Requirements

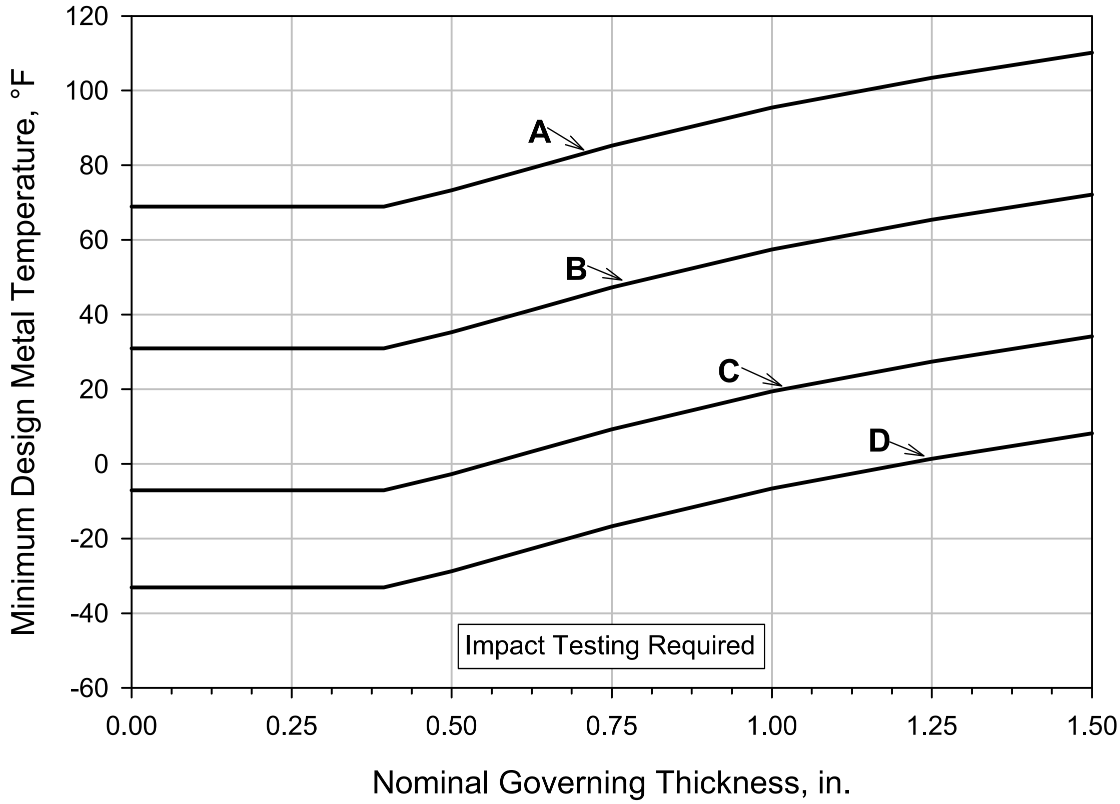


Figure 3.7 – Impact Test Exemption Curves – Parts Not Subject to PWHT

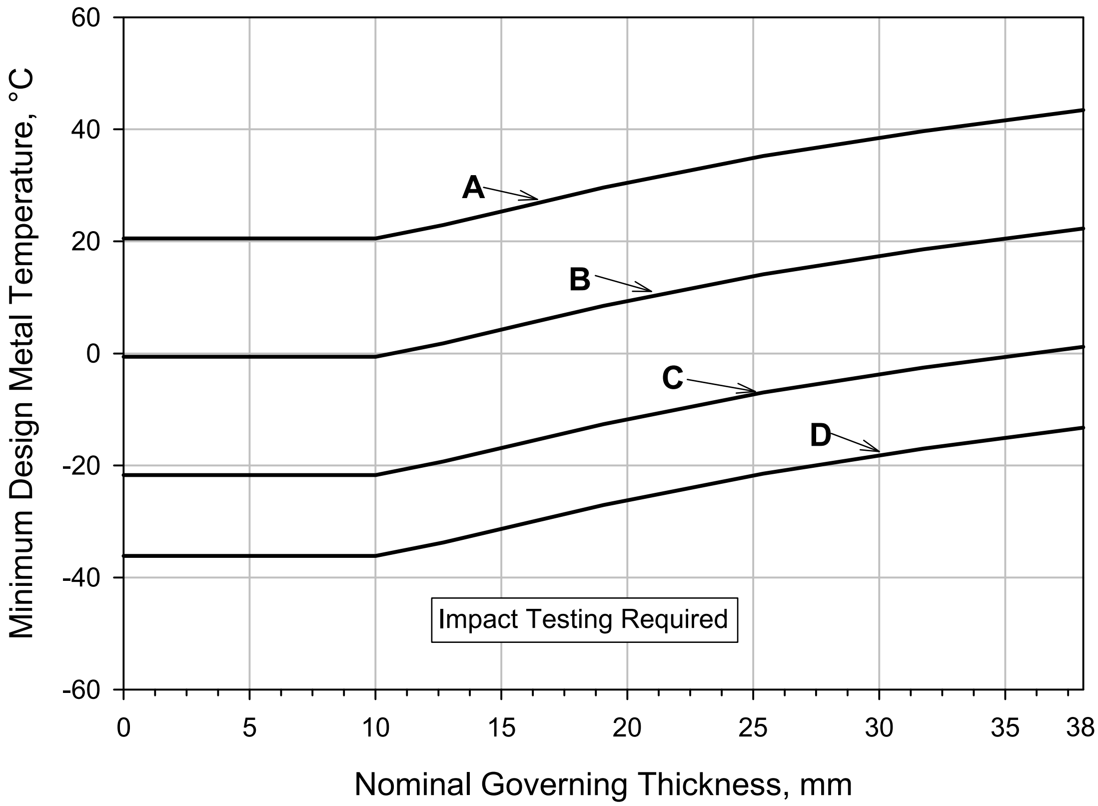


Figure 3.7M – Impact Test Exemption Curves – Parts Not Subject to PWHT

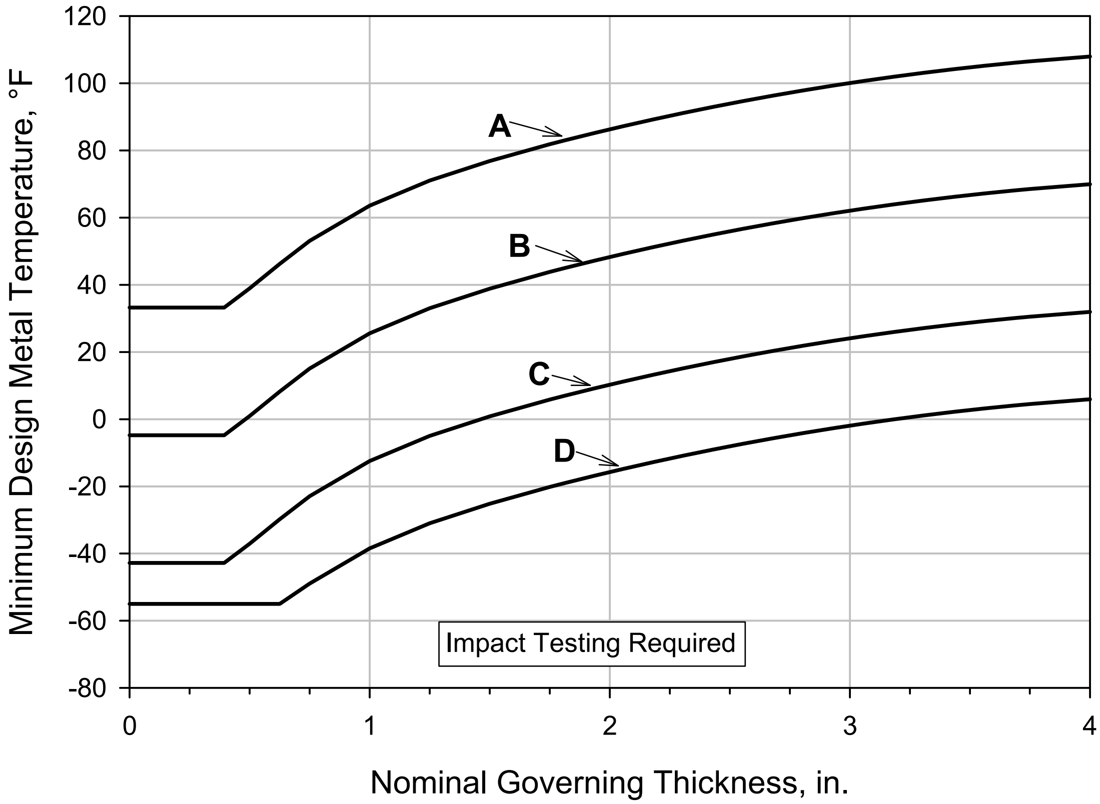


Figure 3.8 – Impact Test Exemption Curves - Parts Subject to PWHT and Non-welded Parts

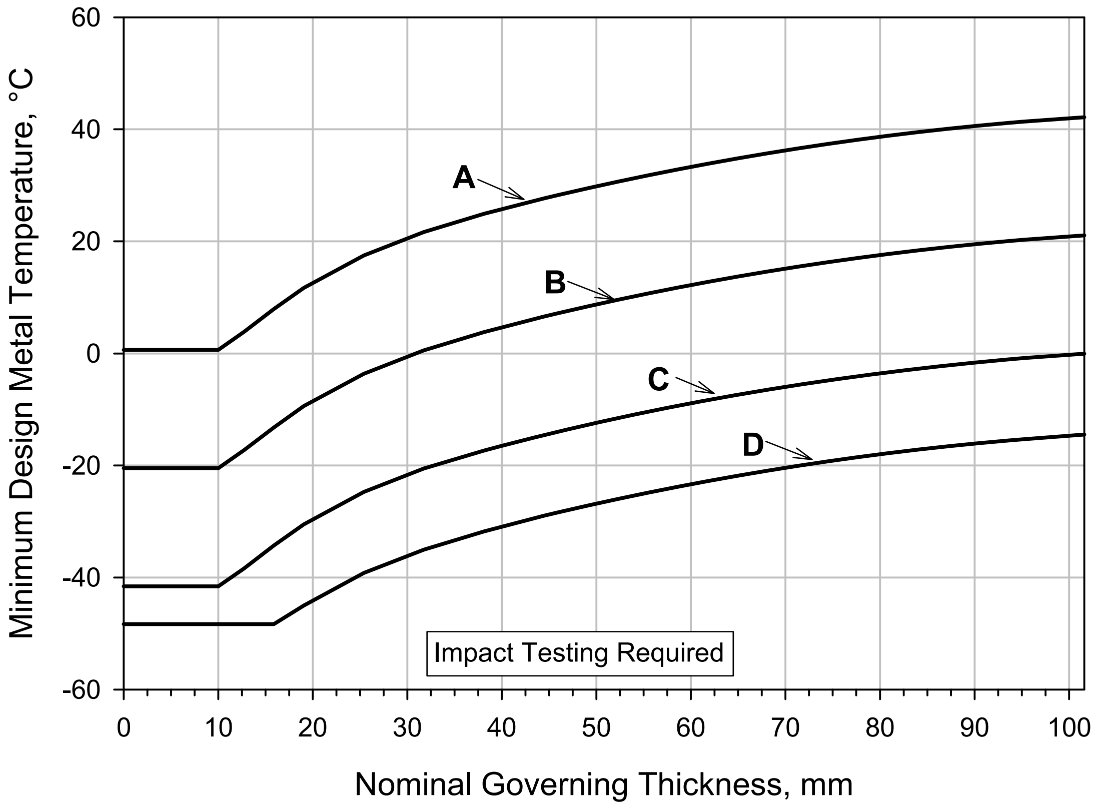


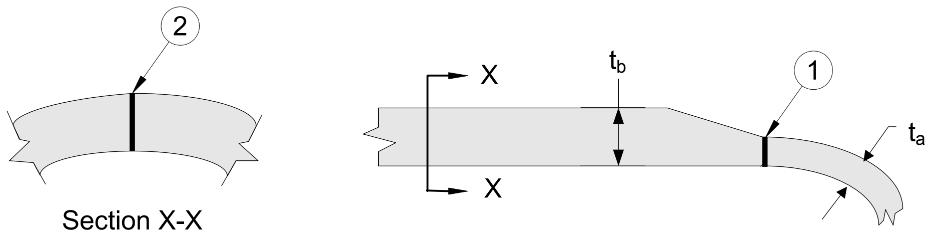
Figure 3.8M – Impact Test Exemption Curves - Parts Subject to PWHT and Non-welded Parts

Material Assignment Table
Based on Exemption Curves and Notes for Figures 3.7, 3.7M, 3.8, and 3.8M

Curve	Material Assignment
A	a) All carbon and all low alloy steel plates, structural shapes and bars not listed in Curves B, C, and D below. b) SA-216 Grades WCB and WCC if normalized and tempered or water-quenched and tempered; SA -217 Grade WC6 if normalized and tempered or water-quenched and tempered
B	a) SA-216 Grades WCA if normalized and tempered or water-quenched and tempered; Grades WCB and WCC for thicknesses not exceeding 50 mm (2 in.) if produced to a fine grain practice and water-quenched and tempered b) SA -217 Grade WC9 if normalized and tempered c) SA-285 Grades A and B d) SA-414 Grade A e) SA-515 Grades 60 f) SA-516 Grades 65 and 70 if not normalized g) SA-662 Grade B if not normalized h) SA/EN 10028-2 Grade P355GH as-rolled i) Except for cast steels, all materials of Curve A if produced to fine grain practice and normalized which are not listed for Curve C and D below; j) Pipe, fittings, forgings, and tubing not listed for Curves C and D below; k) Parts permitted from paragraph 3.2.8, shall be included in Curve B even when fabricated from plate that otherwise would be assigned to a different curve.
C	a) SA-182 Grades F21 and F22 if normalized and tempered. b) SA-302 Grades C and D c) SA-336 Grades F21 and F22 if normalized and tempered, or liquid quenched and tempered. d) SA-387 Grades 21 and 22 if normalized and tempered, or liquid quenched and tempered. e) SA-516 Grades 55 and 60 if not normalized f) SA-533 Grades B and C g) SA-662 Grade A h) All materials listed in (a) through (g) and in (i) for Curve B if produced to fine grain practice and normalized, normalized and tempered, or liquid quenched and tempered as permitted in the material specification, and not listed for Curve D below
D	a) SA-203 b) SA-508 Class 1 c) SA-516 if normalized d) SA-524 Classes 1 and 2 e) SA-537 Classes 1, 2, and 3 f) SA-612 if normalized; except that the increased Cb limit in the footnote of Table 1 of SA-20 is not permitted g) SA-662 if normalized h) SA-738 Grade A i) SA-738 Grade A with Cb and V deliberately added in accordance with the provisions of the material specification, not colder than -29°C (-20°F) j) SA-738 Grade B not colder than -29°C (-20°F) k) SA/EN 10028-2 Grade P355GH if normalized [See Note d)3)]

Notes

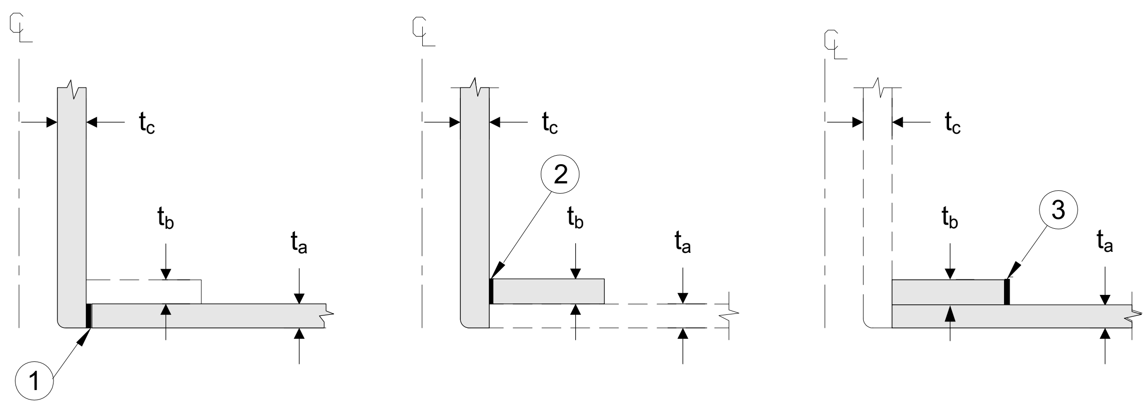
a) Castings not listed as Curve A and B shall be impact tested
b) For bolting see paragraph 3.11.6.
c) When a class or grade is not shown in a material assignment, all classes and grades are indicated.
d) The following apply to all material assignment notes.
   1) Cooling rates faster than those obtained in air, followed by tempering, as permitted by the material specification, are considered equivalent to normalizing and tempering heat treatments.
   2) Fine grain practice is defined as the procedures necessary to obtain a fine austenitic grain size as described in SA-20.
   3) Normalized rolling condition is not considered as being equivalent to normalizing.
e) Data of Figures 3.7 and 3.7M are shown in Table 3.14.
f) Data of Figures 3.8 and 3.8M are shown in Table 3.15.
g) See paragraph 3.11.2.5.a.5.ii for yield strength greater than 450 MPa (65 ksi).



$t_{g1}=t_a$

$t_{g2}=t_a$ (seamless) or t_b (welded)

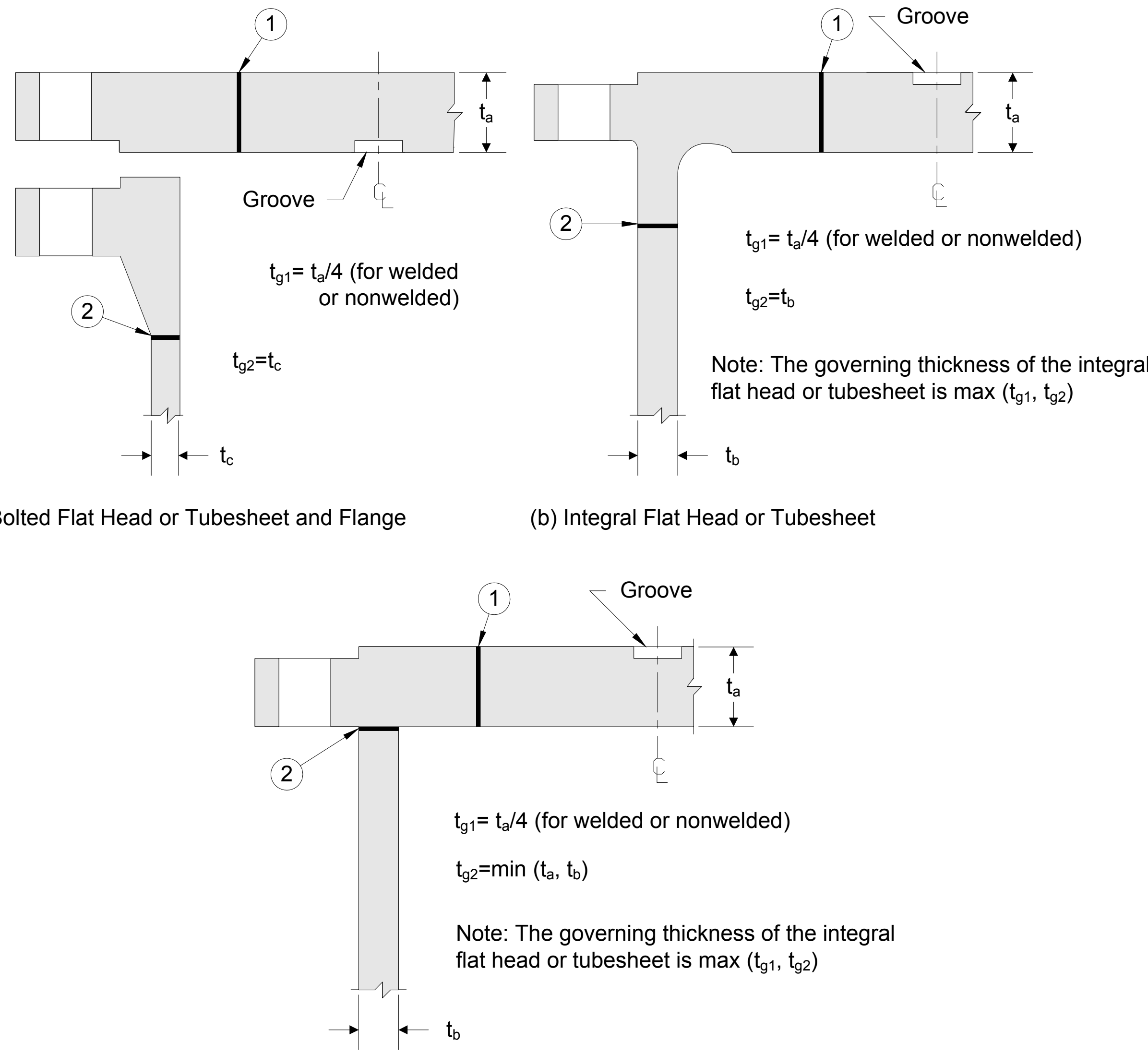
(a) Butt Welded Components



$t_{g1}=\min(t_a, t_c)$ $t_{g2}=\min(t_b, t_c)$ $t_{g3}=\min(t_a, t_b)$

(b) Welded Connection with or without a Reinforcing Plate

Figure 3.9 – Typical Vessel Details Illustrating The Governing Thickness

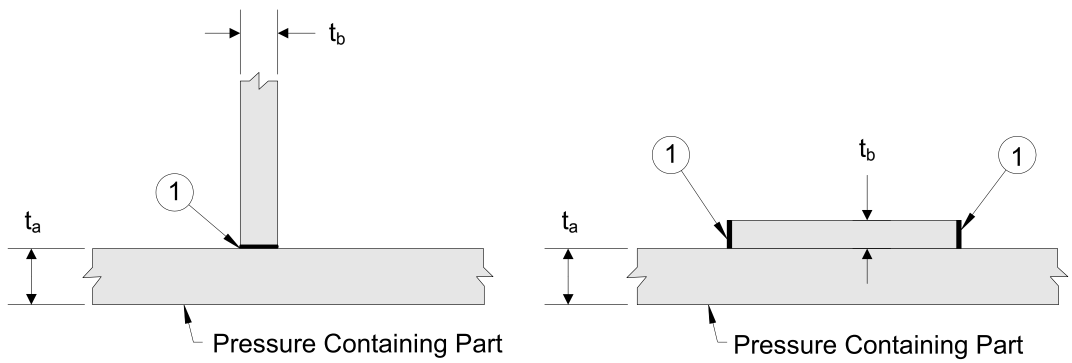


(a) Bolted Flat Head or Tubesheet and Flange

(b) Integral Flat Head or Tubesheet

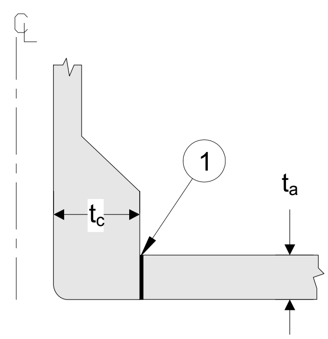
(c) Flat Head or Tubesheet with a Corner Joint

Figure 3.10 – Typical Vessel Details Illustrating the Governing Thickness



t_{g1}=min(t_a, t_b)

(a) Welded Attachments



t_{g1}=min (t_a, t_c)

(b) Integrally Reinforced Welded Connection

Figure 3.11 – Typical Vessel Details Illustrating the Governing Thickness

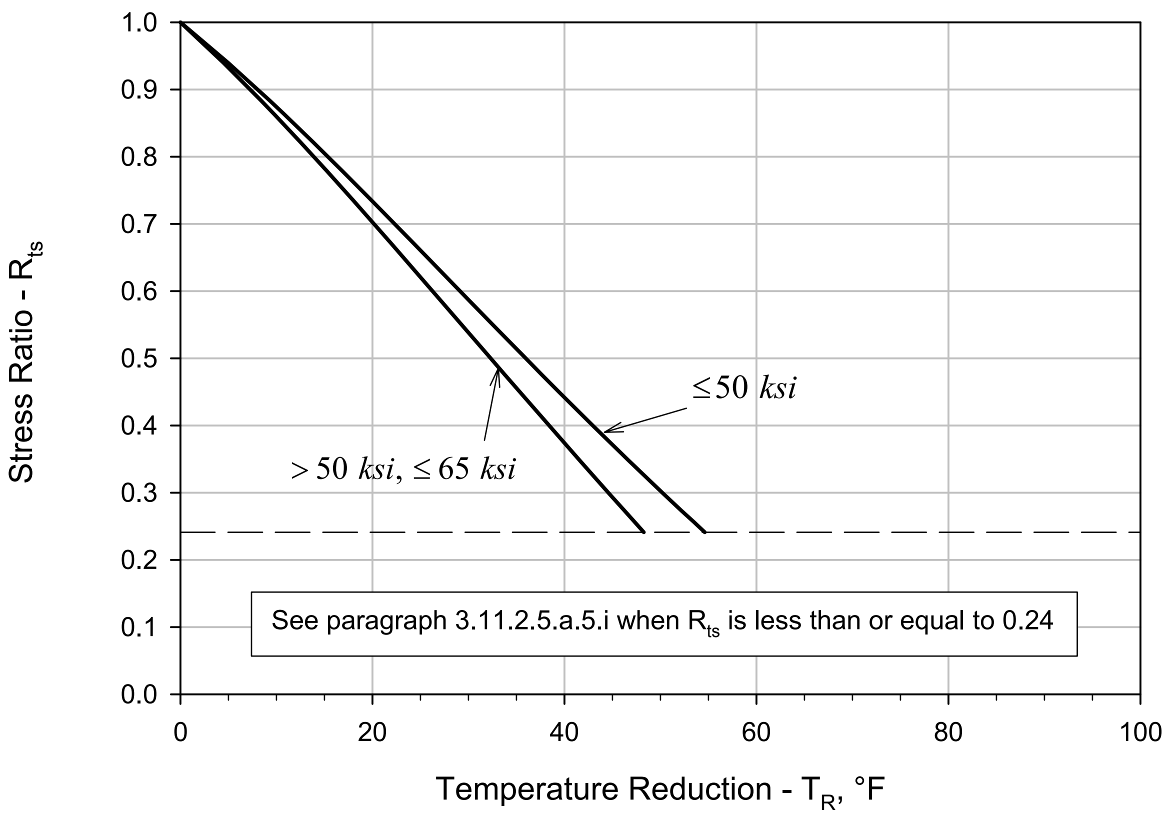


Figure 3.12 – Reduction in the MDMT without Impact Testing – Parts Not Subject to PWHT

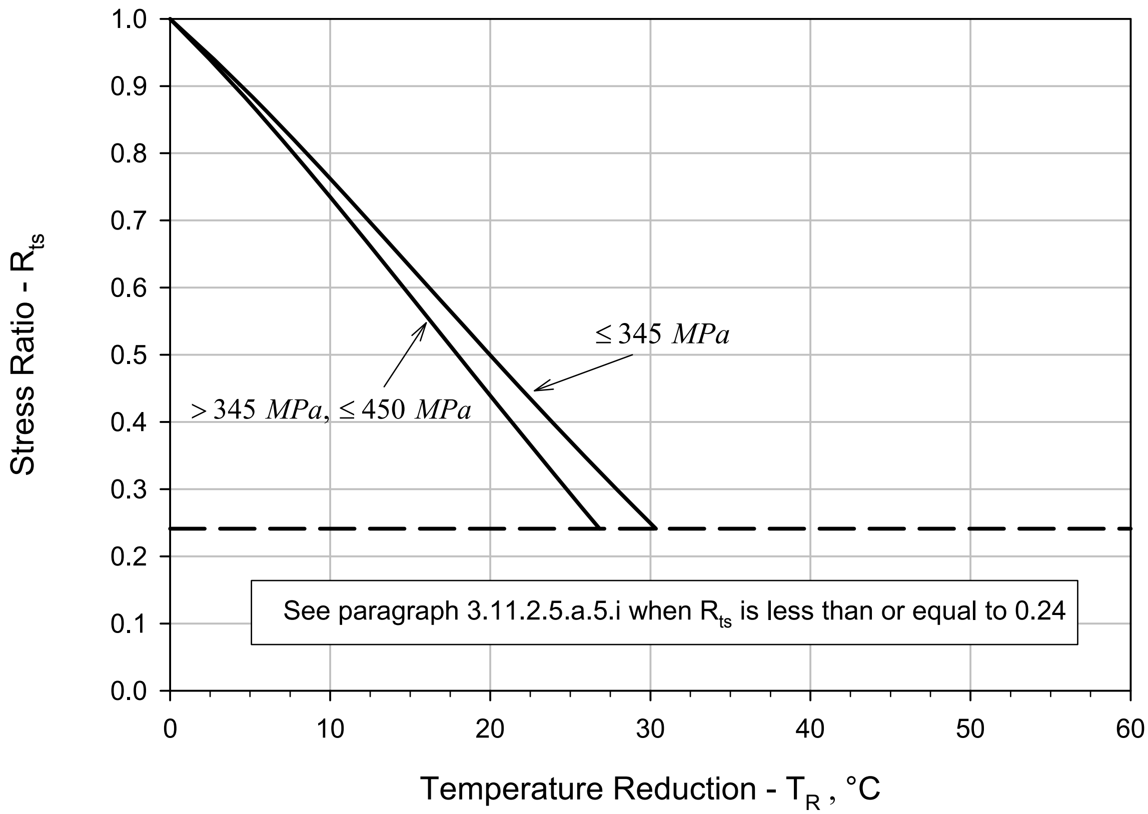


Figure 3.12M – Reduction in the MDMT without Impact Testing – Parts Not Subject to PWHT

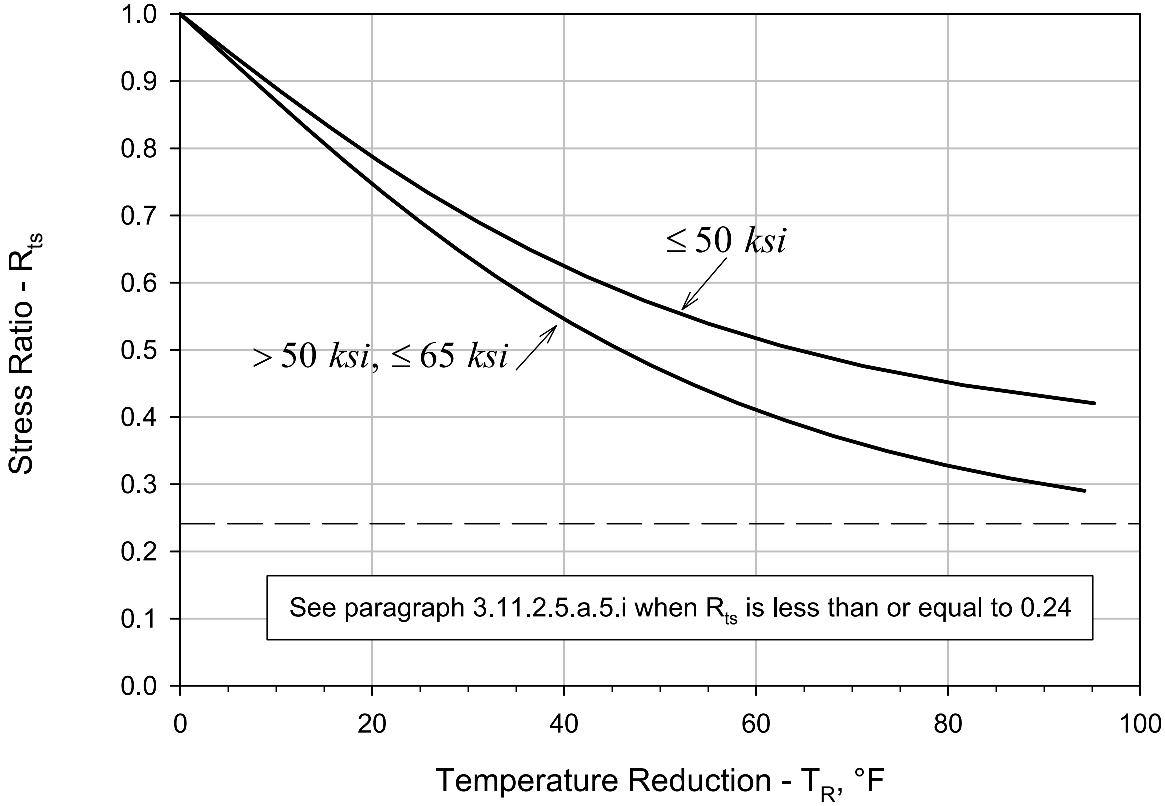


Figure 3.13 – Reduction in the MDMT without Impact Testing - Parts Subject to PWHT and Non-welded Parts

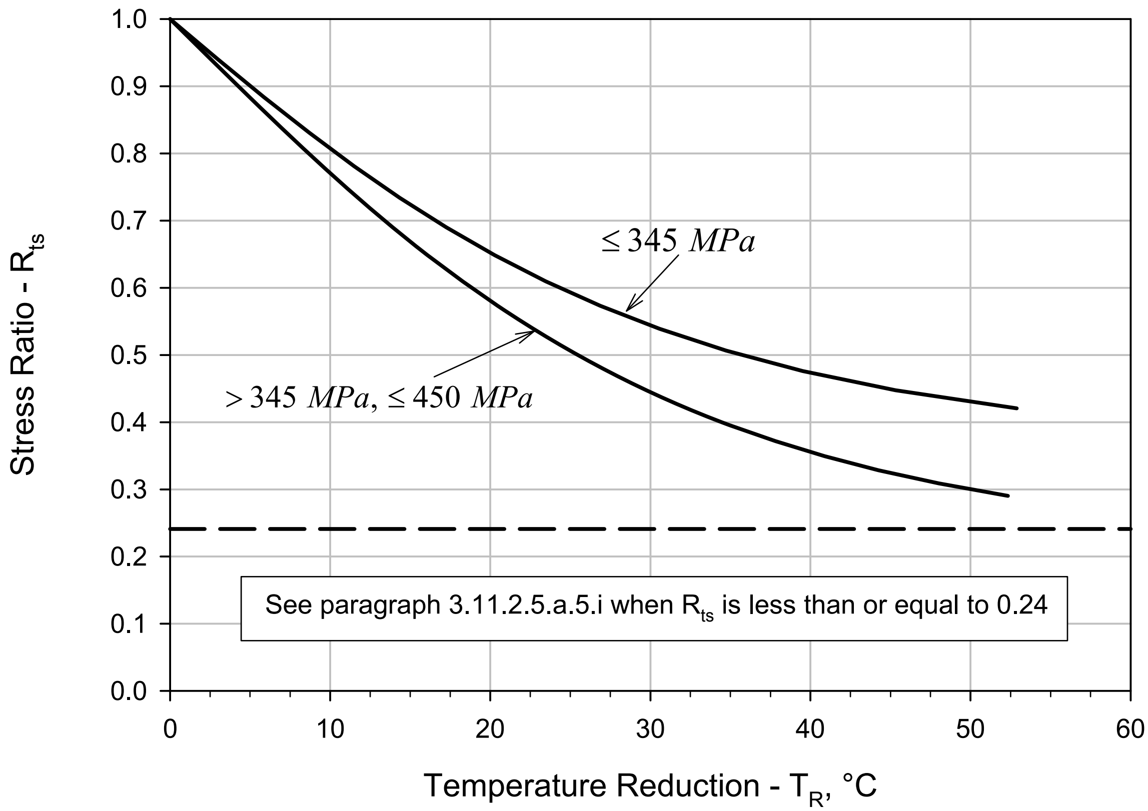
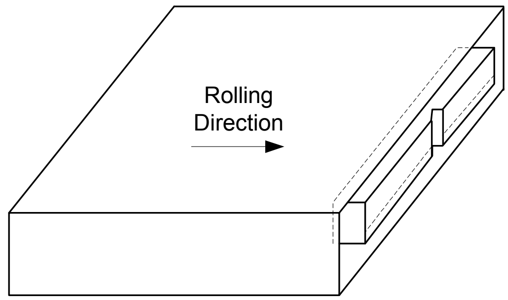


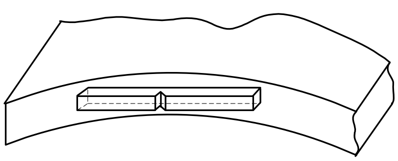
Figure 3.13M – Reduction in the MDMT without Impact Testing - Parts Subject to PWHT and Non-welded Parts

Notes for Figures 3.12, 3.12M, 3.13, and 3.13M

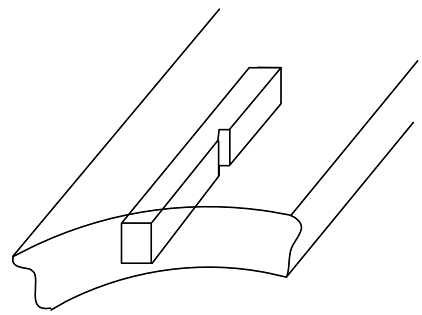
a)	Interpolation between yield strength values is permitted.
b)	The reduction in MDMT shall not exceed 55 ºC (100 ºF), except as permitted by paragraph 3.11.2.5.a.5.i.
c)	Data of Figures 3.12 and 3.12M are shown in Table 3.16.
d)	Data of Figures 3.13 and 3.13M are shown in Table 3.17.



(a) Charpy V-Notch Specimens From Plate

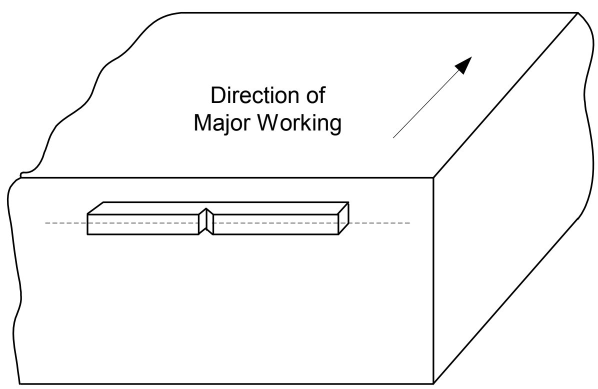


(b-1)



(b-2)

(b) Charpy V-Notch Specimens From Pipe



(c) Charpy V-Notch Specimens From Forgings

Note: The transverse Charpy V-Notch specimen orientation in the pipe shall be as shown in sketch (b-1). If this transverse specimen cannot be obtained from the pipe geometry, then the alternate orientation shown in sketch (b-2) shall be used.

Figure 3.14 – Orientation and Location of Transverse Charpy V-Notch Specimens

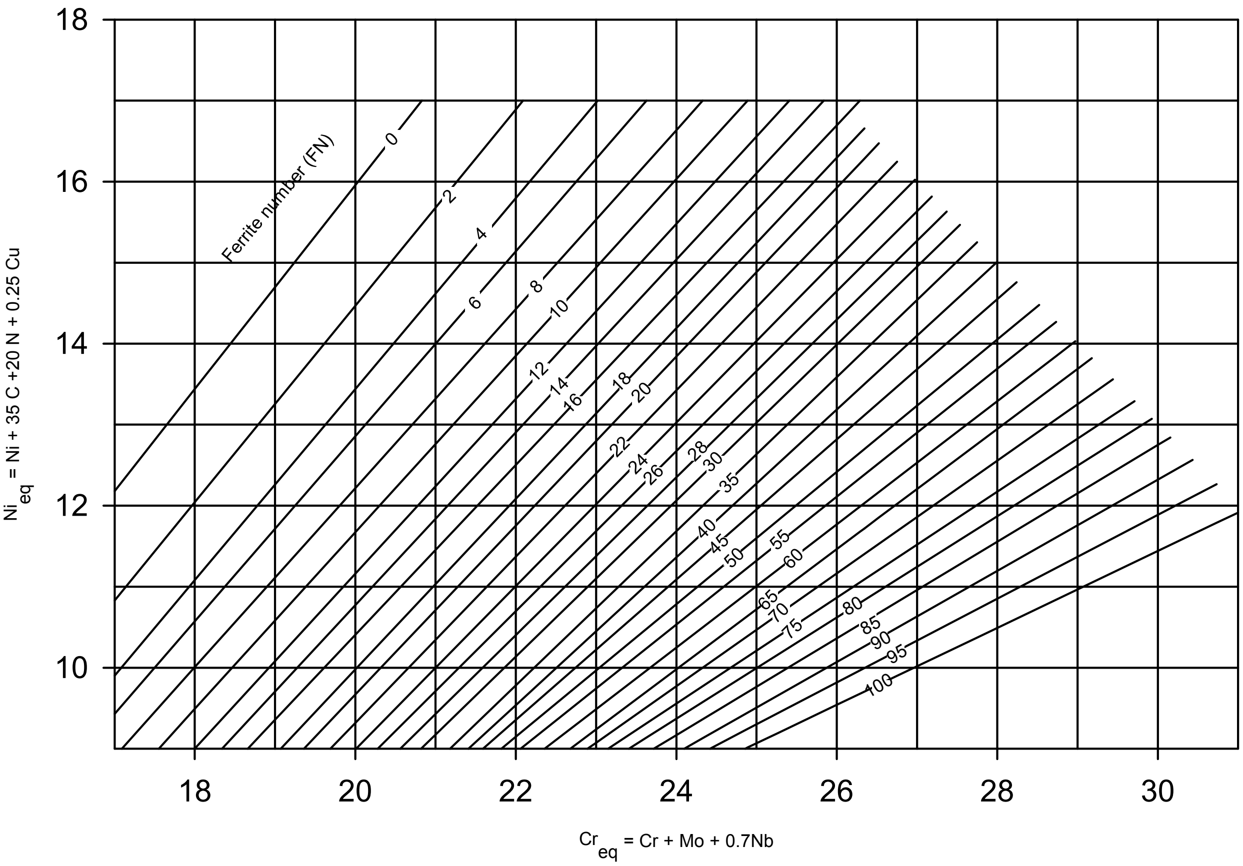


Figure 3.15 – Weld Metal Delta Ferrite Content

ANNEX 3.A

ALLOWABLE DESIGN STRESSES

(NORMATIVE)

3.A.1 Allowable Stress Basis – All Materials Except Bolting

3.A.1.1 The materials that may be used in this Division for all product forms except bolting are shown below.

a) Carbon Steel and Low Alloy Steel – Table 3.A.1

b) Quenched and Tempered High Strength Steels – Table 3.A.2

c) High Alloy Steel – Table 3.A.3

d) Aluminum and Aluminum Alloys – Table 3.A.4

e) Copper and Copper Alloys – Table 3.A.5

f) Nickel and Nickel Alloys – Table 3.A.6

g) Titanium and Titanium Alloys – Table 3.A.7

3.A.1.2 The allowable stresses to be used in this Division for all product forms except bolting are provided in the following tables of Section II, Part D.

a) Carbon Steel and Low Alloy Steel – Section II, Part D, Table 5A

b) Quenched and Tempered High Strength Steels – Section II, Part D, Table 5A

c) High Alloy Steel – Section II, Part D, Table 5A

d) Aluminum and Aluminum Alloys – Section II, Part D, Table 5B

e) Copper and Copper Alloys – Section II, Part D, Table 5B

f) Nickel and Nickel Alloys – Section II, Part D, Table 5B

g) Titanium and Titanium Alloys – Section II, Part D, Table 5B

3.A.2 Allowable Stress Basis – Bolting Materials

3.A.2.1 The materials that may be used in this Division for bolting are shown below.

a) Ferrous Bolting Materials for Design in Accordance With Part 4 of this Division – Table 3.A.8

b) Aluminum Alloy and Copper Alloy Bolting Materials for Design in Accordance With Part 4 of this Division – Table 3.A.9

c) Nickel and Nickel Alloy Bolting Materials Bolting Materials for Design in Accordance With Part 4 of this Division – Table 3.A.10

d) Bolting Materials for Design in Accordance With Part 5 of this Division – Table 3.A.11

3.A.2.2 The allowable stresses to be used in this Division for bolting are provided in the following tables of Section II, Part D.

a) Ferrous Bolting Materials for Design in Accordance With Part 4 of this Division – Section II, Part D, Table 3

b) Aluminum Alloy and Copper Alloy Bolting Materials for Design in Accordance With Part 4 of this Division – Section II, Part D, Table 3

c) Nickel and Nickel Alloy Bolting Materials Bolting Materials for Design in Accordance With Part 4 of this Division – Section II, Part D, Table 3

d) Bolting Materials for Design in Accordance With Part 5 of this Division – Section II, Part D, Table 4

3.A.3 Tables

Table 3.A.1 – Carbon Steel and Low Alloy Materials

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-36	---	K02600	Carbon steel	Str. Plate
SA-105	---	K03504	Carbon steel	Forgings
SA-106	A	K02501	Carbon steel	Smls. Pipe
SA-106	B	K03006	Carbon steel	Smls. Pipe
SA-106	C	K03501	Carbon steel	Smls. Pipe
SA-178	C	K03503	Carbon steel	Wld. Tube
SA-181	60	K03502	Carbon steel	Forgings
SA-181	70	K03502	Carbon steel	Forgings
SA-182	F1	K12822	C-1/2 Mo	Forgings
SA-182	F2	K12122	1/2Cr-1/2Mo	Forgings
SA-182	F3V	K31830	3Cr-1Mo-¼V-Ti-B	Forgings
SA-182	F5	K41545	5Cr-½Mo	Forgings
SA-182	F5a	K42544	5Cr-½Mo	Forgings
SA-182	F9	K90941	9Cr-1Mo	Forgings
SA-182	F12, Cl. 1	K11562	1Cr-1/2Mo	Forgings
SA-182	F12, Cl. 2	K11564	1Cr-1/2Mo	Forgings
SA-182	F11, Cl. 1	K11597	1¼Cr-½Mo-Si	Forgings
SA-182	F11, Cl. 2	K11572	1¼Cr-½Mo-Si	Forgings
SA-182	F21	K31545	3Cr-1Mo	Forgings
SA-182	F22, Cl. 1	K21590	2¼Cr-1Mo	Forgings
SA-182	F22, Cl. 3	K21590	2¼Cr-1Mo	Forgings
SA-182	F22V	K31835	2¼Cr-1Mo-1/4V	Forgings
SA-182	F91	K90901	9Cr-1Mo-V	Forgings
SA-182	FR	K22035	2Ni-1Cu	Forgings
SA-203	A	K21703	2 1/2 Ni	Plate
SA-203	B	K22103	2 1/2 Ni	Plate
SA-203	D	K31718	3 1/2 Ni	Plate
SA-203	E	K32018	3 1/2 Ni	Plate
SA-203	F	---	3 1/2 Ni	Plate
SA-204	A	K11820	C-1/2 Mo	Plate
SA-204	B	K12020	C-1/2 Mo	Plate
SA-204	C	K12320	C-1/2 Mo	Plate
SA-209	T1	K11522	C-1/2 Mo	Smls. Tube
SA-209	T1a	K12023	C-1/2 Mo	Smls. Tube
SA-209	T1b	K11422	C-1/2 Mo	Smls. Tube
SA-210	A-1	K02707	Carbon steel	Smls. Tube
SA-210	C	K03501	Carbon steel	Smls. Tube
SA-213	T2	K11547	½Cr-½Mo	Smls. Tube
SA-213	T5	K41545	5Cr-½ Mo	Smls. Tube
SA-213	T5b	K41545	5Cr-½ Mo-Si	Smls. Tube
SA-213	T5c	K41245	5Cr-½ Mo-Ti	Smls. Tube

Table 3.A.1 – Carbon Steel and Low Alloy Materials

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-213	T9	K90941	9Cr-1Mo	Smls. Tube
SA-213	T11	K11597	1¼Cr-½Mo-Si	Smls. Tube
SA-213	T12	K11562	1Cr-½Mo	Smls. Tube
SA-213	T21	K31545	3Cr-1Mo	Smls. Tube
SA-213	T22	K21590	2½Cr-1Mo	Smls. Tube
SA-213	T91	K90901	9Cr-1Mo-V	Smls. Tube
SA-216	WCA	J02502	Carbon steel	Castings
SA-216	WCB	J03002	Carbon steel	Castings
SA-216	WCC	K02503	Carbon steel	Castings
SA-217	C5	J42045	5Cr-½Mo	Castings
SA-217	C12	J82090	9Cr-1Mo	Castings
SA-217	WC1	J12524	C-½Mo	Castings
SA-217	WC4	J12082	1Ni-½Cr-½Mo	Castings
SA-217	WC5	J22000	¾Ni-1Mo-¾Cr	Castings
SA-217	WC6	J12072	1¼Cr-½Mo	Castings
SA-217	WC9	J21890	2¼Cr-1Mo	Castings
SA-225	C	K12524	Mn-½Ni-V	Plate
SA-234	WPB	K03006	Carbon steel	Fittings
SA-234	WPC	K03501	Carbon steel	Fittings
SA-234	WP1	K12821	C-½Mo	Fittings
SA-234	WP5	K41515	5Cr-½Mo	Fittings
SA-234	WP9	K90941	9Cr-1Mo	Fittings
SA-234	WP11, Cl. 1	---	1¼Cr-½Mo-Si	Fittings
SA-234	WP12, Cl. 1	K12062	1Cr-½Mo	Fittings
SA-234	WP22, Cl. 1	K21590	2¼Cr-1Mo	Fittings
SA-266	1	K03506	Carbon steel	Forgings
SA-266	2	K03506	Carbon steel	Forgings
SA-266	3	K05001	Carbon steel	Forgings
SA-266	4	K03017	Carbon steel	Forgings
SA-283	B	---	Carbon steel	Str. Plate
SA-283	D	---	Carbon steel	Str. Plate
SA-285	A	K01700	Carbon steel	Plate
SA-285	B	K02200	Carbon steel	Plate
SA-285	C	K02801	Carbon steel	Plate
SA-299	---	K02803	Carbon steel	Plate
SA-302	A	K12021	Mn-½Mo	Plate
SA-302	B	K12022	Mn-½Mo	Plate
SA-302	C	K12039	Mn-½Mo-½Ni	Plate
SA-302	D	K12054	Mn-½Mo-¾Ni	Plate
SA-333	1	K03008	Carbon steel	Smls. pipe
SA-333	3	K31918	3½Ni	Smls. pipe
SA-333	4	K11267	¾Cr-¾Ni-Cu-Al	Smls. pipe
SA-333	6	K03006	Carbon steel	Smls. pipe
SA-333	9	K22035	2Ni-1Cu	Smls. pipe

Table 3.A.1 – Carbon Steel and Low Alloy Materials

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-333	1	K03008	Carbon steel	Wld. Pipe
SA-334	1	K03008	Carbon steel	Wld. Tube
SA-334	1	K03008	Carbon steel	Smls. Tube
SA-334	3	K31918	3½Ni	Smls. Tube
SA-334	9	K22035	2Ni-1Cu	Smls. Tube
SA-335	P1	K11522	C-½Mo	Smls. Pipe
SA-335	P2	K11547	½Cr-½Mo	Smls. Pipe
SA-335	P5	K41545	5Cr-½Mo	Smls. Pipe
SA-335	P5b	K51545	5Cr-½Mo-Si	Smls. Pipe
SA-335	P5c	K41245	5Cr-½Mo-Ti	Smls. Pipe
SA-335	P9	K90941	9Cr-1Mo	Smls. Pipe
SA-335	P11	K11597	1¼Cr-½Mo-Si	Smls. Pipe
SA-335	P12	K11562	1Cr-½Mo	Smls. Pipe
SA-335	P21	K31545	3Cr-1Mo	Smls. Pipe
SA-335	P22	K21590	2¼Cr-1Mo	Smls. Pipe
SA-335	P91	K90901	9Cr-1Mo-V	Smls. Pipe
SA-336	F1	K11564	1Cr-½Mo	Forgings
SA-336	F3V	K31830	3Cr-1Mo-¼V-Ti-B	Forgings
SA-336	F5	K41545	5Cr-1Mo	Forgings
SA-336	F5A	K42544	5Cr-1Mo	Forgings
SA-336	F9	K90941	9Cr-1Mo	Forgings
SA-336	F11, Cl. 2	K11572	1¼Cr-½Mo-Si	Forgings
SA-336	F11, Cl. 3	K11572	1¼Cr-½Mo-Si	Forgings
SA-336	F12	K11564	1Cr-½Mo	Forgings
SA-336	F21, Cl. 1	K31545	3Cr-1Mo	Forgings
SA-336	F21, Cl. 3	K31545	3Cr-1Mo	Forgings
SA-336	F22, Cl. 1	K21590	2¼Cr-1Mo	Forgings
SA-336	F22, Cl. 3	K21590	2¼Cr-1Mo	Forgings
SA-336	F22V	K31835	2¼Cr-1Mo-1/4V	Forgings
SA-350	LF1	K03009	Carbon steel	Forgings
SA-350	LF2	K03011	Carbon steel	Forgings
SA-350	LF3	K32025	3½Ni	Forgings
SA-350	LF9	K22036	2Ni-1Cu	Forgings
SA-352	LCB	J03003	Carbon steel	Castings
SA-352	LC1	J12522	C-½Mo	Castings
SA-352	LC2	J22500	2%Ni	Castings
SA-352	LC3	J31550	3½Ni	Castings
SA-369	FP1	K11522	C-½Mo	Forged pipe
SA-369	FP2	K11547	½Cr-½Mo	Forged pipe
SA-369	FP5	K41545	5Cr-½Mo	Forged pipe
SA-369	FP9	K90941	9Cr-1Mo	Forged pipe
SA-369	FP11	K11597	1¼Cr-½Mo-Si	Forged pipe
SA-369	FP12	K11562	1Cr-½Mo	Forged pipe
SA-369	FP21	K31545	3Cr-1Mo	Forged pipe

Table 3.A.1 – Carbon Steel and Low Alloy Materials

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-369	FP22	K21590	2¼Cr-1Mo	Forged pipe
SA-372	A	K03002	Carbon steel	Forgings
SA-372	B	K04001	Carbon steel	Forgings
SA-372	C	K04801	Carbon steel	Forgings
SA-372	D	K10508	Mn-¼Mo	Forgings
SA-387	2, Cl. 1	K12143	½Cr-½Mo	Plate
SA-387	2, Cl. 2	K12143	½Cr-½Mo	Plate
SA-387	5, Cl. 1	K41545	5Cr-½Mo	Plate
SA-387	5, Cl. 2	K41545	5Cr-½Mo	Plate
SA-387	11, Cl. 1	K11789	1¼Cr-½Mo-Si	Plate
SA-387	11, Cl. 2	K11789	1¼Cr-½Mo-Si	Plate
SA-387	12, Cl. 1	K11757	1Cr-½Mo	Plate
SA-387	12, Cl. 2	K11757	1Cr-½Mo	Plate
SA-387	21, Cl. 1	K31545	3Cr-1Mo	Plate
SA-387	21, Cl. 2	K31545	3Cr-1Mo	Plate
SA-387	22, Cl. 1	K21590	2¼Cr-1Mo	Plate
SA-387	22, Cl. 2	K21590	2¼Cr-1Mo	Plate
SA-387	91	K90901	9Cr-1Mo-V	Plate
SA-420	WPL3	---	3½Ni	Fittings
SA-420	WPL6	---	Carbon steel	Fittings
SA-420	WPL9	K22035	2Ni-Cu	Fittings
SA-423	1	K11535	¾Cr-½Ni-Cu	Smls. Tube
SA-423	2	K11540	¾Ni-½Cu-Mo	Smls. Tube
SA-487	1, Cl. A	J13002	Mn-V	Castings
SA-487	4, Cl. A	J13047	½Ni-½Cr-¼Mo-V	Castings
SA-487	8, Cl. A	J22091	2½Cr-1Mo	Castings
SA-508	1	K13502	Carbon steel	Forgings
SA-508	1A	K13502	Carbon steel	Forgings
SA-508	2, Cl. 1	K12766	¾Ni-½Mo-1/3Cr-V	Forgings
SA-508	2, Cl. 2	K12766	¾Ni-½Mo-1/3Cr-V	Forgings
SA-508	3, Cl. 1	K12042	¾Ni-½Mo-Cr-V	Forgings
SA-508	3, Cl. 2	K12042	¾Ni-½Mo-Cr-V	Forgings
SA-508	3V	K31830	3Cr-1Mo-¼V-Ti-B	Forgings
SA-508	4N, Cl. 3	K22375	3½Ni-1¾Cr-½Mo-V	Forgings
SA-508	22, Cl. 3	K215909	2¼Cr-1Mo	Forgings
SA-515	60	K02401	Carbon steel	Plate
SA-515	65	K02800	Carbon steel	Plate
SA-515	70	K03101	Carbon steel	Plate
SA-516	55	K01800	Carbon steel	Plate
SA-516	60	K02100	Carbon steel	Plate
SA-516	65	K02403	Carbon steel	Plate
SA-516	70	K02700	Carbon steel	Plate
SA-524	I	K02104	Carbon steel	Smls. Pipe
SA-524	II	K02104	Carbon steel	Smls. Pipe

Table 3.A.1 – Carbon Steel and Low Alloy Materials

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-533	A, Cl. 1	K12521	Mn-½Mo	Plate
SA-533	A, Cl. 2	K12521	Mn-½Mo	Plate
SA-533	B, Cl. 1	K12539	Mn-½Mo-½Ni	Plate
SA-533	B, Cl. 2	K12539	Mn-½Mo-½Ni	Plate
SA-533	C, Cl. 1	K12554	Mn-½Mo-¾Ni	Plate
SA-533	C, Cl. 2	K12554	Mn-½Mo-¾Ni	Plate
SA-533	D, Cl. 2	K12529	Mn-½Mo-¼Ni	Plate
SA-537	Cl. 1	K12437	Carbon steel	Plate
SA-537	Cl. 2	K12437	Carbon steel	Plate
SA-537	Cl. 3	K12437	Carbon steel	Plate
SA-541	1	K03506	Carbon steel	Forgings
SA-541	1A	K03020	Carbon steel	Forgings
SA-541	2, Cl. 1	K12765	¾Ni-½Mo-1/3Cr-V	Forgings
SA-541	2, Cl. 2	K12765	¾Ni-½Mo-1/3Cr-V	Forgings
SA-541	3, Cl. 1	K12045	½Ni-½Mo-V	Forgings
SA-541	3, Cl. 2	K12045	½Ni-½Mo-V	Forgings
SA-541	3V	K31830	3Cr-1Mo-¼V-Ti-B	Forgings
SA-541	22, Cl. 3	K21390	2¼Cr-1Mo	Forgings
SA-541	22V	K31835	2¼Cr-1Mo-¼V	Forgings
SA-542	B, Cl. 4	---	2¼Cr-1Mo	Plate
SA-542	C, Cl. 4a	---	2¼Cr-1Mo-¼V	Plate
SA-542	D, Cl. 4a	---	3Cr-1Mo-¼V-Ti-B	Plate
SA-612	---	K02900	Carbon steel	Plate
SA-662	A	K10701	Carbon steel	Plate
SA-662	B	K02203	Carbon steel	Plate
SA-662	C	K02007	Carbon steel	Plate
SA-675	45	---	Carbon steel	Bar, shapes
SA-675	50	---	Carbon steel	Bar, shapes
SA-675	55	---	Carbon steel	Bar, shapes
SA-675	60	---	Carbon steel	Bar, shapes
SA-675	65	---	Carbon steel	Bar, shapes
SA-675	70	---	Carbon steel	Bar, shapes
SA-727	---	K02506	Carbon steel	Forgings
SA-737	B	K12001	C-Mn-Si-Cb	Plate
SA-737	C	K12202	C-Mn-Si-V	Plate
SA-738	A	K12447	Carbon steel	Plate
SA-738	B	K12007	Carbon steel	Plate
SA-738	C	---	Carbon steel	Plate
SA-739	B11	K11797	1¼Cr-½Mo	Bar
SA-739	B22	K21390	2¼Cr-1Mo	Bar
SA-765	I	K03046	Carbon steel	Forgings
SA-765	II	K03047	Carbon steel	Forgings
SA-765	III	K32026	3½Ni	Forgings
SA-765	IV	K02009	Carbon steel	Forgings

Table 3.A.1 – Carbon Steel and Low Alloy Materials

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-832	21V	K31830	3Cr-1Mo-¼V-Ti-B	Plate
SA-832	22V	K31835	2¼Cr-1Mo-V	Plate
SA/EN 10028-2	P355GH	---	Carbon Steel	Plate

Table 3.A.2 – Quenched and Tempered High Strength Steels

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-333	8	K81340	9Ni	Smls. Pipe
SA-333	8	K81340	9Ni	Smls. Pipe
SA-334	8	K81340	9Ni	Smls. Tube
SA-334	8	K81340	9Ni	Smls. Tube
SA-353	---	K81340	9Ni	Plate
SA-372	D	K14508	Mn-¼Mo	Forgings
SA-372	E, Cl. 70	K13047	1Cr-1/5Mo	Forgings
SA-372	F, Cl. 70	G41350	1Cr-1/5Mo	Forgings
SA-372	G, Cl. 70	K13049	½Cr-1/5Mo	Forgings
SA-372	H, Cl. 70	K13547	½Cr-1/5Mo	Forgings
SA-372	J, Cl. 70	K13548	1Cr-1/5Mo	Forgings
SA-372	J, Cl. 110	G41370	1Cr-1/5Mo	Forgings
SA-420	WPL8	K81340	9Ni	Smls. Pipe
SA-508	4N, Cl. 1	K22375	3½Ni-1¾Cr-½Mo-V	Forgings
SA-508	4N, Cl. 2	K22375	3½Ni-1¾Cr-½Mo-V	Forgings
SA-517	A	K11856	½Cr-¼Mo-Si	Plate
SA-517	B	K11630	½Cr-1/5Mo-V	Plate
SA-517	E	K21604	1¾Cr-½Mo-Ti	Plate
SA-517	F	K11576	¾Ni-½Cr-½Mo-V	Plate
SA-517	J	K11625	C-½Mo	Plate
SA-517	P	K21650	1¼Ni-1Cr-½Mo	Plate
SA-522	I	K81340	9Ni	Forgings
SA-533	B, Cl. 3	K12554	Mn-½Mo-¾Ni	Plate
SA-533	D, Cl. 3	K12529	Mn-½Mo-¼Ni	Plate
SA-543	B, Cl. 1	K42339	3Ni-1¾Cr-½Mo	Plate
SA-543	B, Cl. 2	K42339	3Ni-1¾Cr-½Mo	Plate
SA-543	B, Cl. 3	K42339	3Ni-1¾Cr-½Mo	Plate
SA-543	C, Cl. 1	---	2¾Ni-1½Cr-½Mo	Plate
SA-543	C, CL. 2	---	2¾Ni-1½Cr-½Mo	Plate
SA-543	C, CL. 3	---	2¾Ni-1½Cr-½Mo	Plate
SA-553	I	K81340	9Ni	Plate
SA-592	A	K11856	½Cr-¼Mo-Si	Forgings
SA-592	E	K11695	1¾Cr-½Mo-Cu	Forgings
SA-592	F	K11576	¾Ni-½Cr-½Mo-V	Forgings
SA-645	A	K41583	5Ni-¼Mo	Plate
SA-723	1, Cl. 1	K23550	2Ni-1½Cr-¼Mo-V	Forgings
SA-723	1, Cl. 2	K23550	2Ni-1½Cr-¼Mo-V	Forgings

Table 3.A.2 – Quenched and Tempered High Strength Steels

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-723	2, Cl. 1	K34035	2¾Ni-1½Cr-½Mo-V	Forgings
SA-723	2, Cl. 2	K34035	2¾Ni-1½Cr-½Mo-V	Forgings
SA-723	3, Cl. 1	K44045	4Ni-1½Cr-½Mo-V	Forgings
SA-723	3, Cl. 2	K44045	4Ni-1½Cr-½Mo-V	Forgings
SA-724	A	K11831	Carbon steel	Plate
SA-724	B	K12031	Carbon steel	Plate
SA-724	C	K12037	Carbon steel	Plate

Table 3.A.3 – High Alloy Steel

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA -182	FXM-11	S21904	21Cr-6Ni-9Mn	Forgings
SA -182	FXM-19	S20910	22Cr-13Ni-5Mn	Forgings
SA -182	F6a, Cl. 1	S41000	13Cr	Forgings
SA -182	F6a, Cl. 2	S41000	13Cr	Forgings
SA -182	F51	S31803	22Cr-5Ni-3Mo-N	Forgings
SA -182	F304	S30400	18Cr-8Ni	Forgings
SA -182	F304H	S30409	18Cr-8Ni	Forgings
SA -182	F304L	S30403	18Cr-8Ni	Forgings
SA -182	F310	S31000	25Cr-20Ni	Forgings
SA -182	F310MoLN	S31050	25Cr-22Ni-2Mo-N	Forgings
SA -182	F316	S31600	16Cr-12Ni-2Mo	Forgings
SA -182	F316H	S31609	16Cr-12Ni-2Mo	Forgings
SA -182	F316L	S31603	16Cr-12Ni-2Mo	Forgings
SA -182	F321	S32100	18Cr-10Ni-Ti	Forgings
SA -182	F321H	S32109	18Cr-10Ni-Ti	Forgings
SA -182	F347	S34700	18Cr-10Ni-Cb	Forgings
SA -182	F347H	S34909	18Cr-10Ni-Cb	Forgings
SA -182	F348	S34800	18Cr-10Ni-Cb	Forgings
SA -182	F348H	S34809	18Cr-10Ni-Cb	Forgings
SA-213	TP304	S30400	18Cr-8Ni	Smls. Tube
SA-213	TP304H	S30409	18Cr-8Ni	Smls. Tube
SA-213	TP304L	S30403	18Cr-8Ni	Smls. Tube
SA-213	TP304N	S30451	18Cr-8Ni-N	Smls. Tube
SA-213	TP309Cb	S30940	23Cr-12Ni-Cb	Smls. Tube
SA-213	TP309H	S30909	23Cr-12Ni	Smls. Tube
SA-213	TP309S	S30908	23Cr-12Ni	Smls. Tube
SA-213	TP310H	S31009	25Cr-20Ni	Smls. Tube
SA-213	TP310MoLN	S31050	25Cr-22Ni-2Mo-N	Smls. Tube
SA-213	TP310S	S31008	25Cr-20Ni	Smls. Tube
SA-213	TP316	S31600	16Cr-12Ni-2Mo	Smls. Tube
SA-213	TP316H	S31609	16Cr-12Ni-2Mo	Smls. Tube
SA-213	TP316L	S31603	16Cr-12Ni-2Mo	Smls. Tube
SA-213	TP316N	S31651	16Cr-12Ni-2Mo-N	Smls. Tube
SA-213	TP321	S32100	18Cr-10Ni-Ti	Smls. Tube
SA-213	TP321H	S32109	18Cr-10Ni-Ti	Smls. Tube
SA-213	TP347	S34700	18Cr-10Ni-Cb	Smls. Tube
SA-213	TP347H	S34709	18Cr-10Ni-Cb	Smls. Tube
SA-213	TP348	S34800	18Cr-10Ni-Cb	Smls. Tube
SA-213	TP348H	S34809	18Cr-10Ni-Cb	Smls. Tube
SA-213	XM-15	S38100	18Cr-18Ni-2Si	Smls. Tube
SA -217	CA15	J91150	13Cr	Castings
SA-240	XM-15	S38100	18Cr-18Ni-2Si	Plate
SA-240	XM19	S20910	22Cr-13Ni-5Mn	Plate

Table 3.A.3 – High Alloy Steel

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-240	XM-29	S24000	18Cr-3Ni-12Mn	Plate
SA-240	XM-29	S24000	18Cr-3Ni-12Mn	Sheet & Strip
SA-240	201LN	S20153	16Cr-4Ni-6Mn	Plate
SA-240	255	S32550	25Cr-5Ni-3Mo-2Cu	Plate
SA-240	302	S30200	18Cr-8Ni	Plate
SA-240	304	S30400	18Cr-8Ni	Plate
SA-240	304H	S30409	18Cr-8Ni	Plate
SA-240	304L	S30403	18Cr-8Ni	Plate
SA-240	304N	S30451	18Cr-8Ni-N	Plate
SA-240	…	S30601	17.5Cr-17.5Ni-5.3Si	Plate
SA-240	309Cb	S30940	23Cr-12Ni-Cb	Plate
SA-240	309H	S30909	23Cr-12Ni	Plate
SA-240	309S	S30908	23Cr-12Ni	Plate
SA-240	310H	S31009	25Cr-20Ni	Plate
SA-240	310MoLN	S31050	25Cr-22Ni-2Mo-N	Plate
SA-240	310S	S31008	25Cr-20Ni	Plate
SA-240	316	S31600	16Cr-12Ni-2Mo	Plate
SA-240	316L	S31603	16Cr-12Ni-2Mo	Plate
SA-240	316N	S31651	16Cr-12Ni-2Mo-N	Plate
SA-240	317	S31700	18Cr-13Ni-3Mo	Plate
SA-240	317L	S31703	18Cr-13Ni-3Mo	Plate
SA-240	321	S32100	18Cr010Ni-Ti	Plate
SA-240	321H	S32109	18Cr-10Ni-Ti	Plate
SA-240	347	S34700	18Cr-10Ni-Cb	Plate
SA-240	347H	S34709	18Cr-10Ni-Cb	Plate
SA-240	348	S34800	18Cr-10Ni-Cb	Plate
SA-240	405	S40500	13Cr-1Al	Plate
SA-240	410	S41000	13 Cr	Plate
SA-240	410S	S41008	13 Cr	Plate
SA-240	429	S42900	15Cr	Plate
SA-240	430	S43000	17Cr	Plate
SA-240	…	S31803	22Cr-5Ni-3Mo-N	Plate
SA-240	26-3-3	S44660	26Cr-3Ni-3Mo	Plate
SA-249	TPXM-15	S38100	18Cr-18Ni-2Si	Wld. Tube
SA-249	TPXM-19	S20910	22Cr-13Ni-5Mn	Wld. Tube
SA-249	TP304	S30400	18Cr-8Ni	Wld. Tube
SA-249	TP304H	S30409	18Cr-8Ni	Wld. Tube
SA-249	TP304L	S30403	18Cr-8Ni	Wld. Tube
SA-249	TP304N	S30451	18Cr-8Ni-N	Wld. Tube
SA-249	TP309Cb	S30940	23Cr-12Ni-Cb	Wld. Tube
SA-249	TP309H	S30909	23Cr-12Ni	Wld. Tube
SA-249	TP309S	S30908	23Cr-12Ni	Wld. Tube
SA-249	TP310Cb	S31040	25Cr-20Ni-Cb	Wld. Tube
SA-249	TP310H	S31009	23Cr-12Ni	Wld. Tube

Table 3.A.3 – High Alloy Steel

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-249	TP310MoLN	S31050	25Cr-22Ni-2Mo-N	Wld. Tube
SA-249	TP310S	S31008	23Cr-12Ni	Wld. Tube
SA-249	TP316	S31600	16Cr-12Ni-2Mo	Wld. Tube
SA-249	TP316H	S31609	16Cr-12Ni-2Mo	Wld. Tube
SA-249	TP316L	S31603	16Cr-12Ni-2Mo	Wld. Tube
SA-249	TP316N	S31651	16Cr-12Ni-2Mo-N	Wld. Tube
SA-249	TP317	S31700	18Cr-3Ni-3Mo	Wld. Tube
SA-249	TP321	S32100	18Cr-10Ni-Ti	Wld. Tube
SA-249	TP321H	S32109	18Cr-10Ni-Ti	Wld. Tube
SA-249	TP347	S34700	18Cr-10Ni-Cb	Wld. Tube
SA-249	TP347H	S34709	18Cr-10Ni-Cb	Wld. Tube
SA-249	TP348	S34800	18Cr-10Ni-Cb	Wld. Tube
SA-249	TP348H	S34809	18Cr-10Ni-Cb	Wld. Tube
SA-268	TP405	S40500	12Cr-1Al	Smls. Tube
SA-268	TP410	S41000	13Cr	Smls. Tube
SA-268	TP429	S42900	15 Cr	Smls. Tube
SA-268	TP430	S43000	17Cr	Smls. Tube
SA-268	26-3-3	S44660	26Cr-3Ni-3Mo	Smls. Tube
SA-268	26-3-3	S44660	26Cr-3Ni-3Mo	Wld. Tube
SA-312	TPXM-11	S21904	21Cr-6Ni-9Mn	Smls. Pipe
SA-312	TPXM-15	S38100	18Cr-18Ni-2Si	Smls. Pipe
SA-312	TPXM-19	S20910	22Cr-13Ni-5Mn	Smls. Pipe
SA-312	TP304	S30400	18Cr-8Ni	Smls. Pipe
SA-312	TP304H	S30409	18Cr-8Ni	Smls. Pipe
SA-312	TP304L	S30403	18Cr-8Ni	Smls. Pipe
SA-312	TP304N	S30451	18Cr-8Ni-N	Smls. Pipe
SA-312	TP309Cb	S30940	23Cr-12Ni-Cb	Smls. Pipe
SA-312	TP309H	S30909	23Cr-12Ni	Smls. Pipe
SA-312	TP309S	S30908	23Cr-12Ni	Smls. Pipe
SA-312	TP310H	S31009	23Cr-12Ni	Smls. Pipe
SA-312	TP310S	S31008	23Cr-12Ni	Smls. Pipe
SA-312	TP316	S31600	16Cr-12Ni-2Mo	Smls. Pipe
SA-312	TP316H	S31609	16Cr-12Ni-2Mo	Smls. Pipe
SA-312	TP316L	S31603	16Cr-12Ni-2Mo	Smls. Pipe
SA-312	TP316N	S31651	16Cr-12Ni-2Mo-N	Smls. Pipe
SA-312	TP317	S31700	18Cr-3Ni-3Mo	Smls. Pipe
SA-312	TP321	S32100	18Cr-10Ni-Ti	Smls. Pipe
SA-312	TP321	S32100	18Cr-10Ni-Ti	Smls. Pipe
SA-312	TP321H	S32109	18Cr-10Ni-Ti	Smls. Pipe
SA-312	TP321H	S32109	18Cr-10Ni-Ti	Smls. Pipe
SA-312	TP347	S34700	18Cr-10Ni-Cb	Smls. Pipe
SA-312	TP347H	S34709	18Cr-10Ni-Cb	Smls. Pipe
SA-312	TP348	S34800	18Cr-10Ni-Cb	Smls. Pipe
SA-312	TP348H	S34809	18Cr-10Ni-Cb	Smls. Pipe

Table 3.A.3 – High Alloy Steel

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-312	TPXM-11	S21904	21Cr-6Ni-9Mn	Wld. Pipe
SA-312	TPXM-15	S38100	18Cr-18Ni-2Si	Wld. Pipe
SA-312	TPXM-19	S20910	22Cr-13Ni-5Mn	Wld. Pipe
SA-312	TP304	S30400	18Cr-8Ni	Wld. Pipe
SA-312	TP304H	S30409	18Cr-8Ni	Wld. Pipe
SA-312	TP304L	S30403	18Cr-8Ni	Wld. Pipe
SA-312	TP304N	S30451	18Cr-8Ni-N	Wld. Pipe
SA-312	TP309Cb	S30940	23Cr-12Ni-Cb	Wld. Pipe
SA-312	TP309H	S30909	23Cr-12Ni	Wld. Pipe
SA-312	TP309S	S30908	23Cr-12Ni	Wld. Pipe
SA-312	TP310Cb	S31040	25Cr-20Ni-Cb	Wld. Pipe
SA-312	TP310H	S31009	23Cr-12Ni	Wld. Pipe
SA-312	TP310MoLN	S31050	25Cr-22Ni-2Mo-N	Wld. Pipe
SA-312	TP310MoLN	S31050	25Cr-22Ni-2Mo-N	Wld. Pipe
SA-312	TP310S	S31008	23Cr-12Ni	Wld. Pipe
SA-312	TP316	S31600	16Cr-12Ni-2Mo	Wld. pipe
SA-312	TP316H	S31609	16Cr-12Ni-2Mo	Wld. pipe
SA-312	TP316L	S31603	16Cr-12Ni-2Mo	Wld. pipe
SA-312	TP316N	S31651	16Cr-12Ni-2Mo-N	Wld. pipe
SA-312	TP317	S31700	18Cr-3Ni-3Mo	Wld. pipe
SA-312	TP321	S32100	18Cr-10Ni-Ti	Wld. pipe
SA-312	TP321H	S32109	18Cr-10Ni-Ti	Wld. pipe
SA-312	TP347	S34700	18Cr-10Ni-Cb	Wld. pipe
SA-312	TP347H	S34709	18Cr-10Ni-Cb	Wld. pipe
SA-312	TP348	S34800	18Cr-10Ni-Cb	Wld. pipe
SA-312	TP348H	S34809	18Cr-10Ni-Cb	Wld. pipe
SA-336	FXM-11	S21904	21Cr-6Ni-9Mn	Forgings
SA-336	FXM-19	S20910	22Cr-13Ni-5Mn	Forgings
SA-336	F6	S41000	13Cr	Forgings
SA-336	F304	S30400	18Cr-8Ni	Forgings
SA-336	F304H	S30409	18Cr-8Ni	Forgings
SA-336	F304L	S30403	18Cr-8Ni	Forgings
SA-336	F304N	S30451	18Cr-8Ni-N	Forgings
SA-336	F310	S31000	25Cr-20Ni	Forgings
SA-336	F316	S31600	16Cr-12Ni-2Mo	Forgings
SA-336	F316H	S31609	16Cr-12Ni-2Mo	Forgings
SA-336	F316L	S31603	16Cr-12Ni-2Mo	Forgings
SA-336	F316N	S31651	16Cr-12Ni-2Mo-N	Forgings
SA-336	F321	S32100	18Cr-10Ni-Ti	Forgings
SA-336	F321H	S32109	18Cr-10Ni-Ti	Forgings
SA-336	F347	S34700	18Cr-10Ni-Cb	Forgings
SA-336	F347H	S34709	18Cr-10Ni-Cb	Forgings
SA-351	CF3	J92500	18Cr-8Ni	Castings
SA-351	CF8	J92600	18Cr-8Ni	Castings

Table 3.A.3 – High Alloy Steel

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-351	CF8C	J92710	18Cr-10Ni-Cb	Castings
SA-351	CF8M	J92900	18Cr-12Ni-2Mo	Castings
SA-351	CF10	J92590	19Cr-9Ni-0.5Mo	Castings
SA-351	CH8	J93400	25Cr-12Ni	Castings
SA-351	CH20	J93402	25Cr-12Ni	Castings
SA-351	CK20	J94202	25Cr-20Ni	Castings
SA-376	TP304	S30400	18Cr-8Ni	Smls. Pipe
SA-376	TP304H	S30409	18Cr-8Ni	Smls. Pipe
SA-376	TP304N	S30451	18Cr-8Ni-N	Smls. Pipe
SA-376	TP316	S31600	16Cr-12Ni-2Mo	Smls. Pipe
SA-376	TP316H	S31609	16Cr-12Ni-2Mo	Smls. Pipe
SA-376	TP316N	S31651	16Cr-12Ni-2Mo-N	Smls. Pipe
SA-376	TP321	S32100	18Cr-10Ni-Ti	Smls. Pipe
SA-376	TP321	S32100	18Cr-10Ni-Ti	Smls. Pipe
SA-376	TP321H	S32109	18Cr-10Ni-Ti	Smls. Pipe
SA-376	TP321H	S32109	18Cr-10Ni-Ti	Smls. Pipe
SA-376	TP347	S34700	18Cr-10Ni-Cb	Smls. Pipe
SA-376	TP347H	S34709	18Cr-10Ni-Cb	Smls. Pipe
SA-376	TP348	S34800	18Cr-10Ni-Cb	Smls. Pipe
SA-403	XM-19	S20910	22Cr-13Ni-5Mn	Fittings
SA-403	304	S30400	18Cr-8Ni	Fittings
SA-403	304H	S30409	18Cr-8Ni	Fittings
SA-403	304L	S30403	18Cr-8Ni	Fittings
SA-403	304N	S30451	18Cr-8Ni-N	Fittings
SA-403	309	S30900	23Cr-12Ni	Fittings
SA-403	310	S31000	25Cr-20Ni	Fittings
SA-403	316	S31600	16Cr-12Ni-2Mo	Fittings
SA-403	316L	S31603	16Cr-12Ni-2Mo	Fittings
SA-403	316N	S31651	16Cr-12Ni-2Mo-N	Fittings
SA-403	317	S31700	18Cr-13Ni-3Mo	Fittings
SA-403	317L	S31700	18Cr-13Ni-3Mo	Fittings
SA-403	321	S32100	18Cr-10Ni-Ti	Fittings
SA-403	321H	S32109	18Cr-10Ni-Ti	Fittings
SA-403	347	S34700	18Cr-10Ni-Cb	Fittings
SA-403	347H	S34709	18Cr-10Ni-Cb	Fittings
SA-403	348	S34800	18Cr-10Ni-Cb	Fittings
SA-403	348H	S34809	18Cr-10Ni-Cb	Fittings
SA-403	XM-19	S20910	22Cr-13Ni-5Mn	Wld. Fittings
SA-403	304	S30400	18Cr-8Ni	Wld. Fittings
SA-403	304H	S30409	18Cr-8Ni	Wld. Fittings
SA-403	304N	S30451	18Cr-8Ni-N	Wld. Fittings
SA-403	309	S30900	23Cr-12Ni	Wld. Fittings
SA-403	310	S31000	25Cr-20Ni	Wld. Fittings
SA-403	316L	S31603	16Cr-12Ni-2Mo	Wld. Fittings

Table 3.A.3 – High Alloy Steel

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-403	316N	S31651	16Cr-12Ni-2Mo-N	Wld. Fittings
SA-403	321	S32100	18Cr-10Ni-Ti	Wld. Fittings
SA-403	321H	S32109	18Cr-10Ni-Ti	Wld. Fittings
SA-403	347	S34700	18Cr-10Ni-Cb	Wld. Fittings
SA-403	347H	S34709	18Cr-10Ni-Cb	Wld. Fittings
SA-403	348	S34800	18Cr-10Ni-Cb	Wld. Fittings
SA-403	348H	S34809	18Cr-10Ni-Cb	Wld. Fittings
SA-479	XM-19	S20910	22Cr-13Ni-5Mn	Bar
SA-479	309H	S30909	23Cr-12Ni	Bar
SA-564	630	S17400	17Cr-4Ni-4Cu	Bar
SA-666	XM-11	S21904	21Cr-6Ni-9Mn	Plate
SA-688	TP304	S30400	18Cr-8Ni	Wld. Tube
SA-688	TP304L	S30403	18Cr-8Ni	Wld. Tube
SA-688	TP316	S31600	16Cr-12Ni-2Mo	Wld. Tube
SA-688	TP316L	S31603	16Cr-12Ni-2Mo	Wld. Tube
SA-693	630	S17400	17Cr-4Ni-4Cu	Plate, Sheet & Strip
SA-705	630	S17400	17Cr-4Ni-4Cu	Forgings
SA-789	---	S31500	18Cr-5Ni-3Mo-N	Smls. Tube
SA789	…	S31803	22Cr-5Ni-3Mo-N	Smls. Tube
SA-789	---	S31500	18Cr-5Ni-3Mo-N	Wld. Tube
SA-789	---	S31803	22Cr-5Ni-3Mo-N	Wld. Tube
SA-790	---	S31500	18Cr-5Ni-3Mo-N	Smls. Pipe
SA-790	…	S31803	22Cr-5Ni-3Mo-N	Smls. Pipe
SA-790	---	S31500	18Cr-5Ni-3Mo-N	Wld. Pipe
SA-790	…	S31803	22Cr-5Ni-3Mo-N	Wld. Pipe
SA-803	26-3-3	S44660	26Cr-3Ni-3Mo	Wld. Tube
SA-813	TP309Cb	S30940	23Cr-12Ni-Cb	Wld. Pipe
SA-813	TP309S	S30908	23Cr-12Ni	Wld. Pipe
SA-813	TP310Cb	S31040	25Cr-20Ni-Cb	Wld. Pipe
SA-813	TP310S	S31008	25Cr-20Ni	Wld. Pipe
SA-814	TP309Cb	S30940	23Cr-12Ni-Cb	Wld. Pipe
SA-814	TP309S	S30908	23Cr-12Ni	Wld. Pipe
SA-814	TP310Cb	S31040	25Cr-20Ni-Cb	Wld. Pipe
SA-814	TP310S	S31008	25Cr-20Ni	Wld. Pipe
SA-965	FXM-11	S21904	21Cr-6Ni-9Mn	Forgings
SA-965	FXM-19	S20910	22Cr-13Ni-5Mn	Forgings
SA-965	F6	S41000	13Cr	Forgings
SA-965	F304	S30400	18Cr-8Ni	Forgings
SA-965	F304H	S30409	18Cr-8Ni	Forgings
SA-965	F304L	S30403	18Cr-8Ni	Forgings
SA-965	F304N	S30451	18Cr-8Ni-N	Forgings
SA-965	F310	S31000	25Cr-20Ni	Forgings
SA-965	F316	S31600	16Cr-12Ni-2Mo	Forgings
SA-965	F316H	S31609	16Cr-12Ni-2Mo	Forgings

Table 3.A.3 – High Alloy Steel

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-965	F316L	S31603	16Cr-12Ni-2Mo	Forgings
SA-965	F316N	S31651	16Cr-12Ni-2Mo-N	Forgings
SA-965	F321	S32100	18Cr-10Ni-Ti	Forgings
SA-965	F321H	S32109	18Cr-10Ni-Ti	Forgings
SA-965	F347	S34700	18Cr-10Ni-Cb	Forgings
SA-965	F347H	S34909	18Cr-10Ni-Cb	Forgings

Table 3.A.4 – Aluminum Alloys

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-209	3003	A93003	Al-Mn-Cu	Plate, sheet
SB-209	3004	A93004	Al-Mn-Mg	Plate, sheet
SB-209	5052	A95052	Al-2.5Mg	Plate, sheet
SB-209	5083	A95083	Al-4.4Mg-Mn	Plate, sheet
SB-209	5086	A95086	Al-4.0Mg-Mn	Plate, sheet
SB-209	5454	A95454	Al-2.7Mg-Mn	Plate, sheet
SB-209	6061	A96061	Al-Mg-Si-Cu	Plate, sheet
SB-210	Allclad 3003	---	Al-Mn-Cu	Smls. drawn tube
SB-210	3003	A93003	Al-Mn-Cu	Smls. drawn tube
SB-210	6061	A96061	Al-Mg-Si-Cu	Smls. drawn tube
SB-210	6063	A96063	Al-Mg-Si	Smls. drawn tube
SB-221	3003	A93003	Al-Mn-Cu	Bar, rod, shapes
SB-221	5083	A95083	Al-4.4Mg-Mn	Bar, rod, shapes
SB-221	5454	A95454	Al-2.7Mg-Mn	Bar, rod, shapes
SB-221	6061	A96061	Al-Mg-Si-Cu	Bar, rod, shapes
SB-221	6063	A96063	Al-Mg-Si	Bar, rod, shapes
SB-241	Allclad 3003	---	Al-Mn-Cu	Smls. extr. tube
SB-241	3003	A93003	Al-Mn-Cu	Smls. extr. tube
SB-241	3003	A93003	Al-Mn-Cu	Smls. Pipe
SB-241	5083	A95083	Al-4.4Mg-Mn	Smls. extr. tube
SB-241	5454	A95454	Al-2.7Mg-Mn	Smls. extr. tube
SB-241	6061	A96061	Al-Mg-Si-Cu	Smls. extr. tube/pipe
SB-241	6061	A96061	Al-Mg-Si-Cu	Smls. drawn pipe
SB-241	6063	A96063	Al-Mg-Si	Smls. extr. tube/pipe
SB-308	6061	A96061	Al-Mg-Si-Cu	Shapes

Table 3.A.5 – Copper Alloys

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-96	---	C65500	97Cu-3.3Si	Plate, sheet
SB-98	---	C65100	98.5Cu-1.5Si	Rod, bar & shapes
SB-98	---	C65500	97Cu-3Si	Rod, bar & shapes
SB-98	---	C66100	94Cu-3Si	Rod, bar & shapes
SB-111	---	C28000	60Cu-20Zn	Smls. Tube
SB-111	---	C44300	71Cu-28Zn-1Sn-0.06As	Smls. Tube
SB-111	---	C44400	71Cu-28Zn-1Sn-0.06Sb	Smls. Tube
SB-111	---	C44500	71Cu-28Zn-1Sn-0.06P	Smls. Tube
SB-111	---	C60800	95Cu-5Al	Smls. Tube
SB-111	---	C70600	90Cu-10Ni	Cond. Tube
SB-111	---	C71500	70Cu-30Ni	Cond. Tube
SB-169	---	C61400	90Cu-7Al-3Fe	Plate, sheet
SB-171	---	C46400	60Cu-39Zn-Sn	Plate
SB-171	---	C70600	90Cu-10Ni	Plate
SB-171	---	C71500	70Cu-30Ni	Plate
SB-187	---	C10200	99.95Cu-P	Rod & bar
SB-187	---	C11000	99.9Cu	Rod & bar
SB-395	---	C70600	90Cu-10Ni	Smls. U-bend tube
SB-395	---	C71500	70Cu-30Ni	Smls. U-bend tube

Table 3.A.6 –Nickel and Nickel Alloys

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-127	---	N04400	67Ni-30Cu	Plate
SB-160	---	N02200	99Ni	Bar, rod
SB-160	---	N02201	99Ni-Low C	Bar, rod
SB-161	---	N02200	99Ni	Smls. pipe & tube
SB-161	---	N02201	99Ni-Low C	Smls. pipe & tube
SB-162	---	N02200	99Ni	Plate, sheet, strip
SB-162	---	N02201	99Ni-Low C	Plate, sheet, strip
SB-163	---	N02200	99Ni	Smls. Tube
SB-163	---	N02201	99Ni-Low C	Smls. Tube
SB-163	---	N04400	67Ni-30Cu	Smls. Tube
SB-163	---	N06600	72Ni-15Cr-8Fe	Smls. Tube
SB-163	---	N08800	33Ni-42Fe-21Cr	Smls. Tube
SB-163	---	N08810	33Ni-42Fe-21Cr	Smls. Tube
SB-163	---	N08825	42Ni-21.5Cr-3Mo-2.3Cu	Smls. Tube
SB-164	---	N04400	67Ni-30Cu	Bar, rod
SB-164	---	N04405	67Ni-30Cu-S	Bar, rod
SB-165	---	N04400	67Ni-30Cu	Smls. pipe & tube
SB-166	---	N06600	72Ni-15Cr-8Fe	Bar, rod
SB-167	---	N06600	72Ni-15Cr-8Fe	Smls. pipe & tube
SB-168	---	N06600	72Ni-15Cr-8Fe	Plate
SB-333	---	N10001	62Ni-28Mo-5Fe	Plate, strip
SB-333	---	N10665	65Ni-28Mo-2Fe	Plate, strip
SB-335	---	N10001	62Ni-28Mo-5Fe	Rod
SB-335	---	N10665	65Ni-28Mo-2Fe	Rod
SB-366	…	N06022	55Ni-21Cr-13.5Mo	Smls. & wld. fittings
SB-366	…	N06059	59Ni-23Cr-16Mo	Smls. fittings
SB-366	…	N10276	54Ni-16Mo-15Cr	Smls. fittings
SB-366	…	N10665	65Ni-28Mo-2Fe	Smls. fittings
SB-366	…	N06059	59Ni-23Cr-16Mo	Wld. fittings
SB-366	…	N10276	54Ni-16Mo-15Cr	Wld. fittings
SB-366	…	N10665	65Ni-28Mo-2Fe	Wld. fittings
SB-407	---	N08800	33Ni-42Fe-21Cr	Smls. Pipe & tube
SB-407	---	N08810	33Ni-42Fe-21Cr	Smls. Pipe & tube
SB-408	---	N08800	33Ni-42Fe-21Cr	Bar, rod
SB-408	---	N08810	33Ni-42Fe-21Cr	Bar, rod
SB-409	---	N08800	33Ni-42Fe-21Cr	Plate
SB-409	---	N08810	33Ni-42Fe-21Cr	Plate
SB-423	---	N08825	42Ni-21.5Cr-3Mo-2.3Cu	Smls. pipe & tube
SB-424	---	N08825	42Ni-21.5Cr-3Mo-2.3Cu	Plate, sheet, strip
SB-425	---	N08825	42Ni-21.5Cr-3Mo-2.3Cu	Bar, rod
SB-434	---	N10003	70Ni-16Mo-7Cr-5Fe	Plate, sheet, strip
SB-435	---	N06002	47Ni-22Cr-9Mo-18Fe	Sheet
SB-435	---	N06002	47Ni-22Cr-9Mo-18Fe	Plate

Table 3.A.6 –Nickel and Nickel Alloys

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-462	…	N06022	55Ni-21Cr-13.5Mo	Forgings
SB-462	…	N06059	59Ni-23Cr-16Mo	Forgings
SB-462	…	N10276	54Ni-16Mo-15Cr	Forgings
SB-462	…	N10665	65Ni-28Mo-2Fe	Forgings
SB-511	---	N08330	35Ni-19Cr-1.25Si	Bar
SB-514	---	N08800	33Ni-42Fe-21Cr	Welded pipe
SB-514	---	N08810	33Ni-42Fe-21Cr	Welded pipe
SB-515	---	N08800	33Ni-42Fe-21Cr	Welded tube
SB-515	---	N08810	33Ni-42Fe-21Cr	Welded tube
SB-516	---	N06600	72Ni-15Cr-8Fe	Welded tube
SB-517	---	N06600	72Ni-15Cr-8Fe	Welded tube
SB-535	---	N08330	35Ni-19Cr-1 1/4 Si	Smls. & welded pipe
SB-536	---	N08330	35Ni-19Cr-1 1/4 Si	Plate, sheet, strip
SB-564	---	N04400	67Ni-30Cu	Forgings
SB-564	…	N06022	55Ni-21Cr-13.5Mo	Forgings
SB-564	…	N06059	59Ni-23Cr-16Mo	Forgings
SB-564	---	N06600	72Ni-15Cr-8Fe	Forgings
SB-564	---	N08800	33Ni-42Fe-21Cr	Forgings
SB-564	---	N08810	33Ni-42Fe-21Cr	Forgings
SB-572	---	N06002	47Ni-22Cr-9Mo-18Fe	Rod
SB-573	---	N10003	70Ni-16Mo-7Cr-5Fe	Rod
SB-574	…	N06022	55Ni-21Cr-13.5Mo	Rod
SB-574	…	N06059	59Ni-23Cr-16Mo	Rod
SB-574	---	N06455	61Ni-16Mo-16Cr	Rod
SB-574	---	N10276	54Ni-16Mo-15Cr	Rod
SB-575	…	N06022	55Ni-21Cr-13.5Mo	Plate, sheet & strip
SB-575	…	N06059	59Ni-23Cr-16Mo	Plate, sheet & strip
SB-575	---	N06455	61Ni-16Mo-16Cr	Plate, sheet & strip
SB-575	---	N10276	54Ni-16Mo-15Cr	Plate, sheet & strip
SB-581	---	N06007	47Ni-22Cr-19Fe-6Mo	Rod
SB-582	---	N06007	47Ni-22Cr-19Fe-6Mo	Plate, sheet, strip
SB-619	---	N06002	47Ni-22Cr-9Mo-18Fe	Welded pipe
SB-619	---	N06007	47Ni-22Cr-19Fe-6Mo	Welded pipe
SB-619	…	N06022	55Ni-21Cr-13.5Mo	Welded pipe
SB-619	…	N06059	59Ni-23Cr-16Mo	Welded pipe
SB-619	---	N06455	61Ni-16Mo-16Cr	Welded pipe
SB-619	---	N10001	62Ni-28Mo-5Fe	Welded pipe
SB-619	---	N10276	54Ni-16Cr-16Mo-5.5Fe	Welded pipe
SB-619	---	N10665	65Ni-28Mo-2Fe	Welded pipe
SB-622	---	N06002	47Ni-22Cr-9Mo-18Fe	Smls. pipe & tube
SB-622	---	N06007	47Ni-22Cr-19Fe-6Mo	Smls. pipe & tube
SB-622	…	N06022	55Ni-21Cr-13.5Mo	Smls. pipe & tube
SB-622	…	N06059	59Ni-23Cr-16Mo	Smls. pipe & tube
SB-622	---	N06455	61Ni-16Mo-16Cr	Smls. pipe & tube

Table 3.A.6 –Nickel and Nickel Alloys

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-622	---	N10001	62Ni-28Mo-5Fe	Smls. pipe & tube
SB-622	---	N10276	54Ni-16Cr-16Mo-5.5Fe	Smls. pipe & tube
SB-622	---	N10665	65Ni-28Mo-2Fe	Smls. pipe & tube
SB-626	---	N06002	47Ni-22Cr-9Mo-18Fe	Welded tube
SB-626	---	N06007	47Ni-22Cr-19Fe-6Mo	Welded tube
SB-626	…	N06022	55Ni-21Cr-13.5Mo	Welded tube
SB-626	…	N06059	59Ni-23Cr-16Mo	Welded tube
SB-626	---	N06455	61Ni-16Mo-16Cr	Welded tube
SB-626	---	N10001	62Ni-28Mo-5Fe	Welded tube
SB-626	---	N10276	54Ni-16Cr-16Mo-5.5Fe	Welded tube
SB-626	---	N10665	65Ni-28Mo-2Fe	Welded tube

Table 3.A.7 – Titanium and Titanium Alloys

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-265	1	R50250	Ti	Plate, sheet, strip
SB-265	2	R50400	Ti	Plate, sheet, strip
SB-265	3	R50550	Ti	Plate, sheet, strip
SB-265	7	R52400	Ti - Pd	Plate, sheet, strip
SB-265	16	R52402	Ti - Pd	Plate, sheet, strip
SB-265	12	R53400	Ti-0.3Mo-0.8Ni	Plate, sheet, strip
SB-338	1	R50250	Ti	Smls. Tube
SB-338	2	R50400	Ti	Smls. Tube
SB-338	3	R50550	Ti	Smls. Tube
SB-338	7	R52400	Ti - Pd	Smls. Tube
SB-338	16	R52402	Ti - Pd	Smls. Tube
SB-338	12	R53400	Ti-0.3Mo-0.8Ni	Smls. Tube
SB-338	1	R50250	Ti	Wld. Tube
SB-338	2	R50400	Ti	Wld. Tube
SB-338	3	R50550	Ti	Wld. Tube
SB-338	7	R52400	Ti - Pd	Wld. Tube
SB-338	16	R52402	Ti - Pd	Wld. Tube
SB-338	12	R53400	Ti-0.3Mo-0.8Ni	Wld. Tube
SB-348	1	R50250	Ti	Bar, billet
SB-348	2	R50400	Ti	Bar, billet
SB-348	3	R50550	Ti	Bar, billet
SB-348	7	R52400	Ti - Pd	Bar, billet
SB-348	16	R52402	Ti - Pd	Bar, billet
SB-348	12	R53400	Ti-0.3Mo-0.8Ni	Bar, billet
SB-381	F1	R50250	Ti	Forgings
SB-381	F2	R50400	Ti	Forgings
SB-381	F3	R50550	Ti	Forgings
SB-381	F7	R52400	Ti - Pd	Forgings
SB-381	F16	R52402	Ti - Pd	Forgings
SB-381	F12	R53400	Ti-0.3Mo-0.8Ni	Forgings
SB-861	1	R50250	Ti	Smls. Pipe
SB-861	2	R50400	Ti	Smls. Pipe
SB-861	3	R50550	Ti	Smls. Pipe
SB-861	7	R52400	Ti - Pd	Smls. Pipe
SB-861	12	R53400	Ti-0.3Mo-0.8Ni	Smls. Pipe
SB-862	1	R50250	Ti	Wld. Pipe
SB-862	2	R50400	Ti	Wld. Pipe
SB-862	3	R50550	Ti	Wld. Pipe
SB-862	7	R52400	Ti - Pd	Wld. Pipe
SB-862	12	R53400	Ti-0.3Mo-0.8Ni	Wld. Pipe

Table 3.A.8 – Ferrous Bolting Materials for Design in Accordance With Part 4

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
Low Alloy Steel Bolts				
SA-193	B5	K50100	5Cr -½Mo	Bolting
SA-193	B7	G41400	1Cr-1/5Mo	Bolting
SA-193	B7M	G41400	1Cr-1/5Mo	Bolting
SA-193	B16	K14072	1Cr-½Mo-V	Bolting
SA-320	L7	G41400	1Cr-1/5Mo	Bolting
SA-320	L7A	G40370	C-¼Mo	Bolting
SA-320	L7M	G41400	1Cr-1/5Mo	Bolting
SA-320	L43	G43400	1¾Ni-¾Cr-¼Mo	Bolting
SA-325	1	K02706	Carbon steel	Bolting
SA-354	BC	K04100	Carbon steel	Bolting
SA-354	BD	K04100	Carbon steel	Bolting
SA-437	B4B	K91352	12Cr-1Mo-V-W	Bolting
SA-437	B4C	K91352	12Cr-1Mo-V-W	Bolting
SA-449	---	K04200	Carbon steel	Bolting
SA-449	---	K04200	Carbon steel	Bolting
SA-449	---	K04200	Carbon steel	Bolting
SA-508	5, Cl. 2	K42365	3½Ni-1¾Cr-½Mo-V	Bolting
SA-540	B21, Cl. 1	K14073	1Cr-½Mo-V	Bolting
SA-540	B21, Cl. 2	K14073	1Cr-½Mo-V	Bolting
SA-540	B21, Cl. 3	K14073	1Cr-½Mo-V	Bolting
SA-540	B21, Cl. 4	K14073	1Cr-½Mo-V	Bolting
SA-540	B21, Cl. 5	K14073	1Cr-½Mo-V	Bolting
SA-540	B23, Cl. 1	H43400	2Ni-¾Cr-¼Mo	Bolting
SA-540	B23, Cl. 2	H43400	2Ni-¾Cr-¼Mo	Bolting
SA-540	B23, Cl. 3	H43400	2Ni-¾Cr-¼Mo	Bolting
SA-540	B23, Cl. 4	H43400	2Ni-¾Cr-¼Mo	Bolting
SA-540	B23, Cl. 5	H43400	2Ni-¾Cr-¼Mo	Bolting
SA-540	B24, Cl. 1	K24064	2Ni-¾Cr-1/3Mo	Bolting
SA-540	B24, Cl. 2	K24064	2Ni-¾Cr-1/3Mo	Bolting
SA-540	B24, Cl. 3	K24064	2Ni-¾Cr-1/3Mo	Bolting
SA-540	B24, Cl. 4	K24064	2Ni-¾Cr-1/3Mo	Bolting
SA-540	B24, Cl. 5	K24064	2Ni-¾Cr-1/3Mo	Bolting
SA-540	B24V, Cl. 3	K24070	2Ni-¾Cr-1/3Mo-V	Bolting
Low Alloy Steel Nuts				
SA-194	2	---	---	Nuts
SA-194	2H	---	---	Nuts
SA-194	2HM	---	---	Nuts
SA-194	3	---	---	Nuts
SA-194	4	---	---	Nuts
SA-194	7	---	---	Nuts
SA-194	7M	---	---	Nuts
SA-194	16	---	---	Nuts

Table 3.A.8 – Ferrous Bolting Materials for Design in Accordance With Part 4

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-540	B21	---	---	Nuts
SA-540	B23	---	---	Nuts
SA-540	B24	---	---	Nuts
SA-540	B24V	---	---	Nuts
High Alloy Steel Bolts				
SA-193	B6	S41000	13Cr	Bolting
SA-193	B8, Cl. 1	S30400	18Cr-8Ni	Bolting
SA-193	B8, Cl. 2	S30400	18Cr-8Ni	Bolting
SA-193	B8C, Cl. 1	S34700	18Cr-10Ni-Cb	Bolting
SA-193	B8C, Cl. 2	S34700	18Cr-10Ni-Cb	Bolting
SA-193	B8M, Cl. 1	S31600	16Cr-12Ni-2Mo	Bolting
SA-193	B8M2	S31600	16Cr-12Ni-2Mo	Bolting
SA-193	B8M2	S31600	16Cr-12Ni-2Mo	Bolting
SA-193	B8M2	S31600	16Cr-12Ni-2Mo	Bolting
SA-193	B8MNA, Cl. 1A	S31651	16Cr-12Ni-2Mo-N	Bolting
SA-193	B8NA, Cl. 1A	S30451	18Cr-8Ni-N	Bolting
SA-193	B8P, Cl. 1	S30500	18Cr-11Ni	Bolting
SA-193	B8P, Cl. 2	S30500	18Cr-11Ni	Bolting
SA-193	B8S	S21800	18Cr-8Ni-4Si-N	Bolting
SA-193	B8SA	S21800	18Cr-8Ni-4Si-N	Bolting
SA-193	B8T, Cl. 1	S32100	18Cr-10Ni-Ti	Bolting
SA-193	B8T, Cl. 2	S32100	18Cr-10Ni-Ti	Bolting
SA-320	B8, Cl. 1	S30400	18Cr-8Ni	Bolting
SA-320	B8, Cl. 2	S30400	18Cr-8Ni	Bolting
SA-320	B8A, Cl. 1A	S30400	18Cr-8Ni	Bolting
SA-320	B8C, Cl. 1	S34700	18Cr-10Ni-Cb	Bolting
SA-320	B8C, Cl. 2	S34700	18Cr-10Ni-Cb	Bolting
SA-320	B8CA, Cl. 1A	S34700	18Cr-10Ni-Cb	Bolting
SA-320	B8F, Cl. 1	S30323	18Cr-8Ni-S	Bolting
SA-320	B8FA, Cl. 1A	S30323	18Cr-8Ni-S	Bolting
SA-320	B8M, Cl. 1	S31600	16Cr-12Ni-2Mo	Bolting
SA-320	B8M, Cl. 2	S31600	16Cr-12Ni-2Mo	Bolting
SA-320	B8MA, Cl. 1A	S31600	16Cr-12Ni-2Mo	Bolting
SA-320	B8T, Cl. 1	S32100	18Cr-10Ni-Ti	Bolting
SA-320	B8T, CL. 2	S32100	18Cr-10Ni-Ti	Bolting
SA-320	B8TA, Cl. 1A	S32100	18Cr-10Ni-Ti	Bolting
SA-453	651, Cl. A	S63198	19Cr-9Ni-Mo-W	Bolting
SA-453	651, Cl. B	S63198	19Cr-9Ni-Mo-W	Bolting
SA-453	660, Cl. A	S66286	25Ni-15Cr-2Ti	Bolting
SA-453	660, Cl. B	S66286	25Ni-15Cr-2Ti	Bolting
SA-479	XM-19	S20910	22Cr-13Ni-5Mn	Bolting
SA-564	630	S17400	17Cr-4Ni-4Cu	Bolting
SA-705	630	S17400	17Cr-4Ni-4Cu	Bolting

Table 3.A.9 – Aluminum Alloy and Copper Alloy Bolting Materials for Design in Accordance With Part 4

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-211	2014	A92014	---	Bolting
SB-211	2024	A92024	---	Bolting
SB-211	6061	A96061	---	Bolting
SB-98	---	C65100	98.5Cu-1.5Si	Rod
SB-98	---	C65500	97Cu-3Si	Rod
SB-98	---	C66100	94Cu-3Si	Rod
SB-150	---	C61400	90Cu-7Al-3Fe	Bar,
SB-150	---	C61400	90Cu-7Al-3Fe	Rod
SB-150	---	C62300	81Cu-10Al-5Ni-3Fe	Bar
SB-150	---	C63000	81Cu-10Al-5Ni-3Fe	Rod
SB-150	---	C63000	81Cu-10Al-5Ni-3Fe	Bar
SB-150	---	C64200	91Cu-7Al-2Si	Bar
SB-150	---	C64200	91Cu-7Al-2Si	Rod
SB-187	---	C10200	99.95Cu-P	Rod
SB-187	---	C11000	99.9Cu	Rod

Table 3.A.10 – Nickel and Nickel Alloy Bolting Materials Bolting Materials for Design in Accordance With Part 4

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SB-160	---	N02200	99 Ni	Bolting
SB-160	---	N02201	99Ni-Low C	Bolting
SB-164		N04400	67Ni-30Cu	Bolting
SB-164	---	N04405	67Ni-30Cu	Bolting
SB-166	---	N06600	72Ni-15Cr-8Fe	Bolting
SB-335	---	N10001	62Ni-28Mo-5Fe	Bolting
SB-335	---	N10665	65Ni-28Mo-2Fe	Bolting
SB-408	---	N08800	33Ni-42Fe-21Cr	Bolting
SB-408	---	N08810	33Ni-42Fe-21Cr	Bolting
SB-425		N08825	42Ni-21.5Cr-3Mo-2.3Cu	Bolting
SB-446	1	N06625	60Ni-22Cr-9Mo-3.5Cb	Bolting
SB-572	---	N06002	47Ni-22Cr-9Mo-18Fe	Bolting
SB-572	---	R30556	21Ni-30Fe-22Cr-18Co-3Mo-3W	Bolting
SB-573	---	N10003	70Ni-16Mo-7Cr-5Fe	Bolting
SB-574	---	N06022	55Ni-21Cr-13.5Mo	Bolting
SB-574	---	N06455	61Ni-16Mo-16Cr	Bolting
SB-574	---	N10276	54Ni-16Mo-15Cr	Bolting
SB-581	---	N06007	47Ni-22Cr-19Fe-6Mo	Bolting
SB-581	---	N06030	40Ni-29Cr-15Fe-5Mo	Bolting
SB-581	---	N06975	49Ni-25Cr-18Fe-6Mo	Bolting
SB-621	---	N08320	26Ni-43Fe-22Cr-5Mo	Bolting
SB-637	---	N07718	53Ni-19Cr-19Fe-Cb-Mo	Bolting
SB-637	2	N07750	70Ni-16Cr-7Fe-Ti-Al	Bolting

Table 3.A.11 – Bolting Materials for Design in Accordance With Part 5

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
Low Alloy Steel Bolts				
SA-193	B5	K50100	5Cr -½Mo	Bolting
SA-193	B7	G41400	1Cr-1/5Mo	Bolting
SA-193	B7M	G41400	1Cr-1/5Mo	Bolting
SA-193	B16	K14072	1Cr-½Mo-V	Bolting
SA-320	L43	G43400	1¾Ni-¾Cr-¼Mo	Bolting
SA-437	B4B	K91352	12Cr-1Mo-V-W	Bolting
SA-437	B4C	K91352	12Cr-1Mo-V-W	Bolting
SA-540	B21 Cl. 1	K14073	1Cr-½Mo-V	Bolting
SA-540	B21 Cl. 2	K14073	1Cr-½Mo-V	Bolting
SA-540	B21 Cl. 3	K14073	1Cr-½Mo-V	Bolting
SA-540	B21 Cl. 4	K14073	1Cr-½Mo-V	Bolting
SA-540	B21 Cl. 5	K14073	1Cr-½Mo-V	Bolting
SA-540	B22 Cl. 1	K41420	1Cr-1Mn-¼Mo	Bolting
SA-540	B22 Cl. 2	K41420	1Cr-1Mn-¼Mo	Bolting
SA-540	B22 Cl. 3	K41420	1Cr-1Mn-¼Mo	Bolting
SA-540	B22 Cl. 4	K41420	1Cr-1Mn-¼Mo	Bolting
SA-540	B22 Cl. 5	K41420	1Cr-1Mn-¼Mo	Bolting
SA-540	B23 Cl. 1	H43400	2Ni-¾Cr-¼Mo	Bolting
SA-540	B23 Cl. 2	H43400	2Ni-¾Cr-¼Mo	Bolting
SA-540	B23 Cl. 3	H43400	2Ni-¾Cr-¼Mo	Bolting
SA-540	B23 Cl. 4	H43400	2Ni-¾Cr-¼Mo	Bolting
SA-540	B23 Cl. 5	H43400	2Ni-¾Cr-¼Mo	Bolting
SA-540	B24 Cl. 1	K24064	2Ni-¾Cr-1/3Mo	Bolting
SA-540	B24 Cl. 2	K24064	2Ni-¾Cr-1/3Mo	Bolting
SA-540	B24 Cl. 3	K24064	2Ni-¾Cr-1/3Mo	Bolting
SA-540	B24 Cl. 4	K24064	2Ni-¾Cr-1/3Mo	Bolting
SA-540	B24 Cl. 5	K24064	2Ni-¾Cr-1/3Mo	Bolting
SA-540	B24V Cl. 3	K24070	2Ni-¾Cr-1/3Mo-V	Bolting
High Alloy Steel Bolts				
SA-193	B6	S41000	13Cr	Bolting
SA-193	B8 Cl. 1	S30400	18Cr-8Ni	Bolting
SA-193	B8C Cl. 1	S34700	18Cr-10Ni-Cb	Bolting
SA-193	B8M Cl. 1	S31600	16Cr-12Ni-2Mo	Bolting
SA-193	B8MNA Cl. 1A	S31651	16Cr-12Ni-2Mo-N	Bolting
SA-193	B8NA Cl. 1A	S30451	18Cr-8Ni-N	Bolting
SA-193	B8S	S21800	18Cr-8Ni-4Si-N	Bolting
SA-193	B8SA	S21800	18Cr-8Ni-4Si-N	Bolting
SA-193	B8T Cl. 1	S32100	18Cr-10Ni-Ti	Bolting
SA-193	B8R, Cl. 1C	S20910	22Cr-13Ni-5Mn	Bolting
SA-193	B8RA	S20910	22Cr-13Ni-5Mn	Bolting
SA-453	651 Cl. A	S63198	19Cr-9Ni-Mo-W	Bolting
SA-453	651 Cl. B	S63198	19Cr-9Ni-Mo-W	Bolting

Table 3.A.11 – Bolting Materials for Design in Accordance With Part 5

Material Specification	Type/Grade/Class	UNS No.	Nominal Composition	Product Form
SA-453	660 Cl. A	S66286	25Ni-15Cr-2Ti	Bolting
SA-453	660 Cl. B	S66286	25Ni-15Cr-2Ti	Bolting
SA-564	630	S17400	17Cr-4Ni-4Cu	Bolting
SA-564	Temper H1100	S17400	17Cr-4Ni-4Cu	Bolting
SA-705	630	S17400	17Cr-4Ni-4Cu	Bolting
SA-705	Temper H1100	S17400	17Cr-4Ni-4Cu	Bolting
Nickel Alloy Bolts				
SB-164	---	N04400	67Ni-30Cu	Bolting
SB-164	---	N04405	67Ni-30Cu-S	Bolting
SB-637	---	N07718	53Ni-19Cr-19Fe-Cb-Mo	Bolting
SB-637	2	N07750	70Ni-16Cr-7Fe-Ti-Al	Bolting

ANNEX 3.B

REQUIREMENTS FOR MATERIAL PROCUREMENT

(CURRENTLY NOT USED)

ANNEX 3.C

ISO MATERIAL GROUP NUMBERS

(CURRENTLY NOT USED)

ANNEX 3.D

STRENGTH PARAMETERS

(NORMATIVE)

3.D.1 Yield Strength

Values for the yield strength as a function of temperature are provided in Table Y-1 in the ASME B&PV Code, Section II, Part D.

3.D.2 Ultimate Tensile Strength

Values for the ultimate tensile strength as a function of temperature are provided in Table U in the ASME B&PV Code, Section II, Part D.

3.D.3 Stress Strain Curve

The following model for the stress-strain curve shall be used in design calculations where required by this Division when the strain hardening characteristics of the stress-strain curve are to be considered. The yield strength and ultimate tensile strength in paragraphs 3.D.1 and 3.D.2 may be used in this model to determine a stress-strain curve at a specified temperature.

$$\varepsilon_t = \frac{\sigma_t}{E_y} + \gamma_1 + \gamma_2 \tag{3.D.1}$$

where

$$\gamma_1 = \frac{\varepsilon_1}{2}\left(1.0 - \tanh\left[H\right]\right) \tag{3.D.2}$$

$$\gamma_2 = \frac{\varepsilon_2}{2}\left(1.0 + \tanh\left[H\right]\right) \tag{3.D.3}$$

$$\varepsilon_1 = \left(\frac{\sigma_t}{A_1}\right)^{\frac{1}{m_1}} \tag{3.D.4}$$

$$A_1 = \frac{\sigma_{ys}\left(1+\varepsilon_{ys}\right)}{\left(\ln\left[1+\varepsilon_{ys}\right]\right)^{m_1}} \tag{3.D.5}$$

$$m_1 = \frac{\ln\left[R\right] + \left(\varepsilon_p - \varepsilon_{ys}\right)}{\ln\left[\dfrac{\ln\left[1+\varepsilon_p\right]}{\ln\left[1+\varepsilon_{ys}\right]}\right]} \tag{3.D.6}$$

$$\varepsilon_2 = \left(\frac{\sigma_t}{A_2}\right)^{\frac{1}{m_2}} \tag{3.D.7}$$

$$A_2 = \frac{\sigma_{uts}\exp\left[m_2\right]}{m_2^{\ m_2}} \tag{3.D.8}$$

$$H = \frac{2\left[\sigma_t - \left(\sigma_{ys} + K\left\{\sigma_{uts} - \sigma_{ys}\right\}\right)\right]}{K\left(\sigma_{uts} - \sigma_{ys}\right)} \tag{3.D.9}$$

$$R = \frac{\sigma_{ys}}{\sigma_{uts}} \tag{3.D.10}$$

$$\varepsilon_{ys} = 0.002 \tag{3.D.11}$$

$$K = 1.5R^{1.5} - 0.5R^{2.5} - R^{3.5} \tag{3.D.12}$$

The parameters m_2, and ε_p are provided in Table 3.D.1.The development of the stress strain curve should be limited to a value of true ultimate tensile stress at true ultimate tensile strain. The stress strain curve beyond this point should be perfectly plastic. The value of true ultimate tensile stress at true ultimate tensile strain is calculated as follows:

$$\sigma_{uts,t} = \sigma_{uts}\exp\left[m_2\right] \tag{3.D.13}$$

3.D.4 Cyclic Stress Strain Curve

The cyclic stress-strain curve of a material (i.e. strain amplitude versus stress amplitude) may be represented by the Equation (3.D.14). The material constants for this model are provided in Table 3.D.2.

$$\varepsilon_{ta} = \frac{\sigma_a}{E_y} + \left[\frac{\sigma_a}{K_{css}}\right]^{\frac{1}{n_{css}}} \tag{3.D.14}$$

The hysteresis loop stress-strain curve of a material (i.e. strain range versus stress range) obtained by scaling the cyclic stress-strain curve by a factor of two is represented by the Equation (3.D.15). The material constants provided in Table 3.D.2 are also used in this equation.

$$\varepsilon_{tr} = \frac{\sigma_r}{E_y} + 2\left[\frac{\sigma_r}{2K_{css}}\right]^{\frac{1}{n_{css}}} \tag{3.D.15}$$

3.D.5 Tangent Modulus

3.D.5.1 Tangent Modulus Based on the Stress-Strain Curve Model

The tangent modulus based on the stress-strain curve model in paragraph 3.D.3 is given by the following equation.

$$E_t = \frac{\partial \sigma_t}{\partial \varepsilon_t} = \left(\frac{\partial \varepsilon_t}{\partial \sigma_t} \right)^{-1} = \left(\frac{1}{E_y} + D_1 + D_2 + D_3 + D_4 \right)^{-1} \tag{3.D.16}$$

where

$$D_1 = \frac{\sigma_t^{\left(\frac{1}{m_1}-1\right)}}{2m_1 A_1^{\left(\frac{1}{m_1}\right)}} \tag{3.D.17}$$

$$D_2 = -\frac{1}{2}\left(\frac{1}{A_1^{\left(\frac{1}{m_1}\right)}} \right) \cdot \left(\sigma_t^{\left(\frac{1}{m_1}\right)} \left\{ \frac{2}{K\left(\sigma_{uts} - \sigma_{ys}\right)} \right\} \left\{ 1 - \tanh^2\left[H\right] \right\} + \frac{1}{m_1} \sigma_t^{\left(\frac{1}{m_1}-1\right)} \tanh\left[H\right] \right) \tag{3.D.18}$$

$$D_3 = \frac{\sigma_t^{\left(\frac{1}{m_2}-1\right)}}{2m_2 A_2^{\left(\frac{1}{m_2}\right)}} \tag{3.D.19}$$

$$D_4 = \frac{1}{2}\left(\frac{1}{A_2^{\left(\frac{1}{m_2}\right)}} \right) \cdot \left(\sigma_t^{\left(\frac{1}{m_2}\right)} \left\{ \frac{2}{K\left(\sigma_{uts} - \sigma_{ys}\right)} \right\} \left\{ 1 - \tanh^2\left[H\right] \right\} + \frac{1}{m_2} \sigma_t^{\left(\frac{1}{m_2}-1\right)} \tanh\left[H\right] \right) \tag{3.D.20}$$

The parameter K is given by Equation (3.D.12)

3.D.5.2 Tangent Modulus Based on External Pressure Charts

An acceptable alternative for calculating the Tangent Modulus is to use the External Pressure charts in Section II, Part D, Subpart 3, including the notes to Subpart 3. The appropriate chart for the material under consideration is assigned in the column designated External Pressure Chart Number given in Tables 1A or 1B. The tangent modulus, E_t, is equal to $2B/A$, where A is the strain given on the abscissa and B is the stress value on the ordinate of the chart.

3.D.6 Nomenclature

A	Section II, Part D, Subpart 3 external pressure chart A-value.
A_1	curve fitting constant for the elastic region of the stress-strain curve.
A_2	curve fitting constant for the plastic region of the stress-strain curve.
B	Section II, Part D, Subpart 3 external pressure chart B-value.
D_1	coefficient used in the tangent modulus.
D_2	coefficient used in the tangent modulus.
D_3	coefficient used in the tangent modulus.
D_4	coefficient used in the tangent modulus.
ε_p	stress-strain curve fitting parameter.
ε_t	total true strain
ε_{ta}	total true strain amplitude.
ε_{tr}	total true strain range.
ε_{ys}	0.2% engineering offset strain.
ε_1	true plastic strain in the micro-strain region of the stress-strain curve.
ε_2	true plastic strain in the macro-strain region of the stress-strain curve.
E_t	tangent modulus of elasticity evaluated at the temperature of interest.
E_y	modulus of elasticity evaluated at the temperature of interest, see Annex 3.E.
γ_1	true strain in the micro-strain region of the stress-strain curve.
γ_2	true strain in the macro-strain region of the stress-strain curve.
H	stress-strain curve fitting parameter.
K	material parameter for stress-strain curve model
K_{css}	material parameter for the cyclic stress-strain curve model.
m_1	curve fitting exponent for the stress-strain curve equal to the true strain at the proportional limit and the strain hardening coefficient in the large strain region.
m_2	curve fitting exponent for the stress-strain curve equal to the true strain at the true ultimate stress.
n_{css}	material parameter for the cyclic stress-strain curve model.
σ_a	total stress amplitude.
σ_r	total stress range.
σ_t	true stress at which the true strain will be evaluated, may be a membrane, membrane plus bending, or membrane, membrane plus bending plus peak stress depending on the application.
σ_{ys}	engineering yield stress evaluated at the temperature of interest, see paragraph 3.D.1.
σ_{uts}	engineering ultimate tensile stress evaluated at the temperature of interest, see paragraph 3.D.2.
$\sigma_{uts,t}$	true ultimate tensile stress evaluated at the true ultimate tensile strain
R	engineering yield to engineering tensile ratio.

3.D.7 Tables

Table 3.D.1 – Stress-Strain Curve Parameters

Material	Temperature Limit	m_2	ε_p
Ferritic Steel	480°C (900°F)	$0.60(1.00-R)$	2.0E-5
Stainless Steel and Nickel Base Alloys	480°C (900°F)	$0.75(1.00-R)$	2.0E-5
Duplex Stainless Steel	480°C (900°F)	$0.70(0.95-R)$	2.0E-5
Precipitation Hardenable Nickel Base	540°C (1000°F)	$1.90(0.93-R)$	2.0E-5
Aluminum	120°C (250°F)	$0.52(0.98-R)$	5.0E-6
Copper	65°C (150°F)	$0.50(1.00-R)$	5.0E-6
Titanium and Zirconium	260°C (500°F)	$0.50(0.98-R)$	2.0E-5

Table 3.D.2 – Cyclic Stress-Strain Curve Data

Material Description	Temperature (°F)	n_{css}	K_{css} (ksi)
Carbon Steel (0.75 in. – base metal)	70	0.128	109.8
	390	0.134	105.6
	570	0.093	107.5
	750	0.109	96.6
Carbon Steel (0.75 in. – weld metal)	70	0.110	100.8
	390	0.118	99.6
	570	0.066	100.8
	750	0.067	79.6
Carbon Steel (2 in. – base metal)	70	0.126	100.5
	390	0.113	92.2
	570	0.082	107.5
	750	0.101	93.3
Carbon Steel (4 in. – base metal)	70	0.137	111.0
	390	0.156	115.7
	570	0.100	108.5
	750	0.112	96.9
1Cr–1/2Mo (0.75 in. – base metal)	70	0.116	95.7
	390	0.126	95.1
	570	0.094	90.4
	750	0.087	90.8
1Cr–1/2Mo (0.75 in. – weld metal)	70	0.088	96.9
	390	0.114	102.7
	570	0.085	99.1
	750	0.076	86.9
1Cr–1/2Mo (0.75 in. – base metal)	70	0.105	92.5
	390	0.133	99.2
	570	0.086	88.0
	750	0.079	83.7
1Cr–1Mo–1/4V	70	0.128	156.9
	750	0.128	132.3
	930	0.143	118.2
	1020	0.133	100.5
	1110	0.153	80.6
2-1/4Cr–1/2Mo	70	0.100	115.5
	570	0.109	107.5
	750	0.096	105.9
	930	0.105	94.6
	1110	0.082	62.1

Table 3.D.2 – Cyclic Stress-Strain Curve Data

Material Description	Temperature (°F)	n_{css}	K_{css} (ksi)
9Cr–1Mo	70	0.177	141.4
	930	0.132	100.5
	1020	0.142	88.3
	1110	0.121	64.3
	1200	0.125	49.7
Type 304	70	0.171	178.0
	750	0.095	85.6
	930	0.085	79.8
	1110	0.090	65.3
	1290	0.094	44.4
Type 304 (Annealed)	70	0.334	330.0
800H	70	0.070	91.5
	930	0.085	110.5
	1110	0.088	105.7
	1290	0.092	80.2
	1470	0.080	45.7
Aluminum (Al–4.5Zn–0.6Mn)	70	0.058	65.7
Aluminum (Al–4.5Zn–1.5Mg)	70	0.047	74.1
Aluminum (1100-T6)	70	0.144	22.3
Aluminum (2014-T6)	70	0.132	139.7
Aluminum (5086)	70	0.139	96.0
Aluminum (6009-T4)	70	0.124	83.7
Aluminum (6009-T6)	70	0.128	91.8
Copper	70	0.263	99.1

Table 3.D.2M – Cyclic Stress-Strain Curve Data

Material Description	Temperature (C)	n_{css}	K_{css} (MPa)
Carbon Steel (20 mm – base metal)	20	0.128	757
	200	0.134	728
	300	0.093	741
	400	0.109	666
Carbon Steel (20 mm – weld metal)	20	0.110	695
	200	0.118	687
	300	0.066	695
	400	0.067	549
Carbon Steel (50 mm – base metal)	20	0.126	693
	200	0.113	636
	300	0.082	741
	400	0.101	643
Carbon Steel (100 mm – base metal)	20	0.137	765
	200	0.156	798
	300	0.100	748
	400	0.112	668
1Cr–1/2Mo (20 mm – base metal)	20	0.116	660
	200	0.126	656
	300	0.094	623
	400	0.087	626
1Cr–1/2Mo (20 mm – weld metal)	20	0.088	668
	200	0.114	708
	300	0.085	683
	400	0.076	599
1Cr–1/2Mo (50 mm – base metal)	20	0.105	638
	200	0.133	684
	300	0.086	607
	400	0.079	577
1Cr–1Mo–1/4V	20	0.128	1082
	400	0.128	912
	500	0.143	815
	550	0.133	693
	600	0.153	556
2-1/4Cr–1/2Mo	20	0.100	796
	300	0.109	741
	400	0.096	730
	500	0.105	652
	600	0.082	428
9Cr–1Mo	20	0.117	975
	500	0.132	693
	550	0.142	609
	600	0.121	443

Table 3.D.2M – Cyclic Stress-Strain Curve Data

Material Description	Temperature (C)	n_{css}	K_{css} (MPa)
	650	0.125	343
Type 304	20	0.171	1227
	400	0.095	590
	500	0.085	550
	600	0.090	450
	700	0.094	306
Type 304 (Annealed)	20	0.334	2275
800H	20	0.070	631
	500	0.085	762
	600	0.088	729
	700	0.092	553
	800	0.080	315
Aluminum (Al–4.5Zn–0.6Mn)	20	0.058	453
Aluminum (Al–4.5Zn–1.5Mg)	20	0.047	511
Aluminum (1100-T6)	20	0.144	154
Aluminum (2014-T6)	20	0.132	963
Aluminum (5086)	20	0.139	662
Aluminum (6009-T4)	20	0.124	577
Aluminum (6009-T6)	20	0.128	633
Copper	20	0.263	683

ANNEX 3.E

PHYSICAL PROPERTIES

(NORMATIVE)

3.E.1 Young's Modulus

Values for the Young's Modulus as a function of temperature are provided in ASME B&PV Code, Section II, Part D.

3.E.2 Thermal Expansion Coefficient

Values for the thermal expansion coefficient as a function of temperature are provided in ASME B&PV Code, Section II, Part D.

3.E.3 Thermal Conductivity

Values for the thermal conductivity as a function of temperature are provided in ASME B&PV Code, Section II, Part D.

3.E.4 Thermal Diffusivity

Values for the thermal diffusivity as a function of temperature are provided in ASME B&PV Code, Section II, Part D.

ANNEX 3.F
DESIGN FATIGUE CURVES
(NORMATIVE)

3.F.1 Smooth Bar Design Fatigue Curves

3.F.1.1 Smooth bar design fatigue curves in paragraph 3.F.1.1 are provided for the following materials in terms of a polynomial function, see Equation (3.F.1). The constants for these functions, C_n, are provided for different fatigue curves as described below.

a) Carbon, Low Alloy, Series 4xx, and High Tensile Strength Steels for temperatures not exceeding 371°C (700°F) where $\sigma_{uts} \le 552\ MPa\ (80\ ksi)$ (see Table 3.F.1).

b) Carbon, Low Alloy Series 4xx, and High Tensile Strength Steels for temperatures not exceeding 371°C (700°F) where $\sigma_{uts} = 793 - 892\,MPa\ (115 - 130\ ksi)$ (see Table 3.F.2).

c) Series 3xx High Alloy Steels, Nickel-Chromium-Iron Alloy, Nickel-Iron-Chromium Alloy, and Nickel-Copper Alloy for temperatures not exceeding 427°C (800°F) where $S_a > 195\,MPa\ (28.2\ ksi)$ (see Table 3.F.3).

d) Series 3xx High Alloy Steels, Nickel-Chromium-Iron Alloy, Nickel-Iron-Chromium Alloy, and Nickel-Copper Alloy for temperatures not exceeding 427°C (800°F) where $S_a \le 195\,MPa\ (28.2\ ksi)$ (see Table 3.F.4).

e) Wrought 70-30 Copper-Nickel for temperatures not exceeding 232°C (450°F) (see Tables 3.F.5, 3.F.6, and 3.F.7). These data are applicable only for materials with minimum specified yield strength as shown. These data may be interpolated for intermediate values of minimum specified yield strength.

f) Nickel-Chromium-Molybdenum-Iron, Alloys X, G, C-4, And C-276 for temperatures not exceeding 427°C (800°F) (see Table 3.F.8).

g) High strength bolting for temperatures not exceeding 371°C (700°F) (see Table 3.F.9).

3.F.1.2 The design number of design cycles, N, can be computed from Equation (3.F.1) or Table 3.F.10 based on the stress amplitude, S_a, which is determined in accordance with Part 5 of this Division.

$$N = 10^X \tag{3.F.1}$$

where

$$X = \frac{C_1 + C_3 Y + C_5 Y^2 + C_7 Y^3 + C_9 Y^4 + C_{11} Y^5}{1 + C_2 Y + C_4 Y^2 + C_6 Y^3 + C_8 Y^4 + C_{10} Y^5} \tag{3.F.2}$$

$$Y = \left(\frac{S_a}{C_{us}}\right) \cdot \left(\frac{E_{FC}}{E_T}\right) \tag{3.F.3}$$

3.F.2 Welded Joint Design Fatigue Curves

3.F.2.1 Subject to the limitations of paragraph 5.5.5, the welded joint design fatigue curves in paragraph 3.F.2.1 can be used to evaluate welded joints for the following materials and associated temperature limits.

a) Carbon, Low Alloy, Series 4xx, and High Tensile Strength Steels for temperatures not exceeding 371°C (700°F)

b) Series 3xx High Alloy Steels, Nickel-Chromium-Iron Alloy, Nickel-Iron-Chromium Alloy, and Nickel-Copper Alloy for temperatures not exceeding 427°C (800°F)

c) Wrought 70 Copper-Nickel for temperatures not exceeding 232°C (450°F)

d) Nickel-Chromium-Molybdenum-Iron, Alloys X, G, C-4, And C-276 for temperatures not exceeding 427°C (800°F)

e) Aluminum Alloys

3.F.2.2 The number of allowable design cycles for the welded joint fatigue curve shall be computed as follows.

a) The design number of allowable design cycles, N , can be computed from Equation (3.F.4) based on the equivalent structural stress range parameter, $\Delta S_{ess,k}$, determined in accordance with paragraph 5.5.5 of this Division. The constants C and h for use in Equation (3.F.4) are provided in Table 3.F.11. The lower 99% Prediction Interval (-3σ) shall be used for design unless otherwise agreed to by the Owner-User and the Manufacturer.

$$N = \frac{f_I}{f_E}\left(\frac{f_{MT} \cdot C}{\Delta S_{ess,k}}\right)^{\frac{1}{h}} \tag{3.F.4}$$

b) If a fatigue improvement method is performed that exceeds the fabrication requirements of this Division, then a fatigue improvement factor, f_I, may be applied. The fatigue improvement factors shown below may be used. An alternative factor determined may also be used if agreed to by the user or user's designated agent and the Manufacturer.

1) For burr grinding in accordance with Part 6, Figure 6.2.

$$f_I = 1.0 + 2.5 \cdot (10)^q \tag{3.F.5}$$

2) For TIG dressing

$$f_I = 1.0 + 2.5 \cdot (10)^q \tag{3.F.6}$$

3) For hammer peening

$$f_I = 1.0 + 4.0 \cdot (10)^q \tag{3.F.7}$$

In the above equations, the parameter q is given by the following equation.

$$q = -0.0016 \cdot \left(\frac{\Delta S_{ess,k}}{C_{usm}}\right)^{1.6} \tag{3.F.8}$$

c) The design fatigue cycles given by Equation (3.F.4) may be modified to account for the effects of environment other than ambient air that may cause corrosion or sub-critical crack propagation. The environmental modification factor, f_E, is typically a function of the fluid environment, loading frequency, temperature, and material variables such as grain size and chemical composition. A value of $f_E = 4.0$ shall be used unless there is specific information to justify an alternate value based on the severity of the material/environmental interaction. The environmental modification factor, f_E, shall be specified in the User's Design Specification.

d) A temperature adjustment is required to the fatigue curve for materials other than carbon steel and/or for temperatures above 21°C (70°F). The temperature adjustment factor is given by Equation (3.F.9).

$$f_{MT} = \frac{E_T}{E_{ACS}} \tag{3.F.9}$$

3.F.3 Nomenclature

A	constant in the weld joint fatigue curve equation.
B	exponent in the weld joint fatigue curve equation.
$C_1 \rightarrow C_{11}$	equation constants used to represent the smooth bar fatigue curves.
C_{us}	conversion factor, $C_{us} = 1.0$ for units of stress in ksi and $C_{us} = 6.894757$ for units of stress in MPa.
C_{usm}	conversion factor, $C_{usm} = 1.0$ for units of stress in ksi and $C_{usm} = 14.148299$ for units of stress in MPa.
E_{ACS}	modulus of elasticity of carbon steel at ambient temperature or 21°C (70°F).
E_{FC}	modulus of elasticity used to establish the design fatigue curve
E_T	modulus of elasticity of the material under evaluation at the average temperature of the cycle being evaluated.
f_E	environmental correction factor to the welded joint fatigue curve.
f_I	fatigue improvement method correction factor to the welded joint fatigue curve.
f_{MT}	material and temperature correction factor to the welded joint fatigue curve.
q	parameter used to determine the effect equivalent structural stress range on the fatigue improvement factor.
N	number of allowable design cycles.
S_a	computed stress amplitude from Part 5.
$\Delta S_{ess,k}$	computed equivalent structural stress range parameter from Part 5.
σ_{uts}	minimum specified ultimate tensile strength.
X	exponent used to compute the permissible number of cycles.
Y	stress factor used to compute X.

3.F.4 Tables

Table 3.F.1 – Coefficients for Fatigue Curve 110.1 – Carbon, Low Alloy, Series 4XX, High Alloy Steels, And High Tensile Strength Steels For Temperatures not Exceeding 371 °C (700 °F) – $\sigma_{uts} \leq 552\, MPA\ (80\, ksi)$

Coefficients C_i	$48 \leq S_a < 214 (MPa)$ $7 \leq S_a < 31 (ksi)$	$214 \leq S_a \leq 3999 (MPa)$ $31 \leq S_a \leq 580 (ksi)$
1	2.254510E+00	7.999502E+00
2	-4.642236E-01	5.832491E-02
3	-8.312745E-01	1.500851E-01
4	8.634660E-02	1.273659E-04
5	2.020834E-01	-5.263661E-05
6	-6.940535E-03	0.0
7	-2.079726E-02	0.0
8	2.010235E-04	0.0
9	7.137717E-04	0.0
10	0.0	0.0
11	0.0	0.0
Note: $E_{FC} = 195E3\ MPa\ (28.3E3\ ksi)$		

Table 3.F.2 – Coefficients for Fatigue Curve 110.1 – Carbon, Low Alloy, Series 4XX, High Alloy Steels, And High Tensile Strength Steels For Temperatures not Exceeding 371 °C (700 °F) – $\sigma_{uts} = 793 - 892\,MPA\ (115 - 130\,ksi)$

Coefficients C_i	$77.2 \le S_a \le 296(MPa)$ $11.2 \le S_a \le 43(ksi)$	$296 < S_a \le 2896(MPa)$ $43 < S_a \le 420(ksi)$
1	1.608291E+01	8.628486E+00
2	-4.113828E-02	-1.264052E-03
3	-1.023740E+00	-1.605097E-04
4	3.544068E-05	-2.548491E-03
5	2.896256E-02	-1.409031E-02
6	1.826072E-04	8.557033E-05
7	3.863423E-04	5.059948E-04
8	0.0	6.913396E-07
9	0.0	-2.354834E-07
10	0.0	0.0
11	0.0	0.0
Note: $E_{FC} = 195E3\ MPa\ (28.3E3\ ksi)$		

Table 3.F.3 – Coefficients For Fatigue Curve 110.2.1 – Series 3XX High Alloy Steels, Nickel-Chromium-Iron Alloy, Nickel-Iron-Chromium Alloy, And Nickel-Copper Alloy For Temperatures Not Exceeding 427 °C (800 °F) Where $S_a > 195\ MPa\ (28.2\, ksi)$

Coefficients C_i	$195 \le S_a \le 4881 (MPa)$ $28.3 \le S_a \le 708 (ksi)$
1	2.114025E+01
2	1.536993E-01
3	4.487599E-01
4	3.651302E-04
5	-8.981314E-05
6	0.0
7	0.0
8	0.0
9	0.0
10	0.0
11	0.0
Notes: $E_{FC} = 195E3\ MPa\ (28.3E3\ ksi)$	

Table 3.F.4 – Coefficients For Fatigue Curve 110.2.2 – Series 3xx High Alloy Steels, Nickel-Chromium-Iron Alloy, Nickel-Iron-Chromium Alloy, And Nickel-Copper Alloy For Temperatures Not Exceeding 427 °C (800 °F) Where $S_a \le 195\ MPa\ (28.2\, ksi)$

Coefficients C_i	Curve A	Curve B	Curve C
	$163 \le S_a \le 194(MPa)$ $23.7 \le S_a \le 28.2(ksi)$	$114 \le S_a \le 194(MPa)$ $16.5 \le S_a \le 28.2(ksi)$	$94 \le S_a \le 194(MPa)$ $13.6 \le S_a \le 28.2(ksi)$
1	1.785872E+01	5.856886E+00	1.891543E+01
2	-4.973162E-02	-1.774051E-01	1.478023E-01
3	-2.264248E+00	-1.070324E+00	-2.436005E+00
4	-4.356560E-03	1.033285E-02	-6.369796E-02
5	9.567759E-02	6.513423E-02	-6.908655E-02
6	3.327393E-04	-1.879553E-04	5.379540E-03
7	-1.347445E-03	-1.320281E-03	1.957898E-02
8	-5.717269E-06	-5.633339E-07	-1.391592E-04
9	0.0	0.0	-6.506543E-04
10	0.0	0.0	0.0
11	0.0	0.0	0.0

Notes:

1. $E_{FC} = 206E3\ MPa\ (30E3\ ksi)$
2. Thermal bending stresses resulting from either axial or through-wall gradients are not included in Q.
3. Curve A is to be used with inelastic analysis.
4. The maximum effect of retained mean stress is included in Curve C.
5. The criterion to select a curve is shown in the following table.

Curve	Elastic Analysis Of Material Other Than Welds And Heat Effected Zones	Elastic Analysis of Welds and Heat Effected Zones
A	$(P_L + P_b + Q)_{range} \le 188\ MPa\ (27.2\, ksi)$	…
B	$(P_L + P_b + Q)_{range} > 188\ MPa\ (27.2\, ksi)$ and S_a is corrected for applied mean stress	$(P_L + P_b + Q)_{range} \le 188\ MPa\ (27.2\, ksi)$
C	$(P_L + P_b + Q)_{range} > 188\ MPa\ (27.2\, ksi)$ and S_a is not corrected for applied mean stress	$(P_L + P_b + Q)_{range} > 188\ MPa\ (27.2\, ksi)$

Table 3.F.5 – Coefficients for Fatigue Curve 110.3 – Wrought 70 Copper-Nickel For Temperatures Not Exceeding 371 °C (700 °F) – $\sigma_{ys} = 134\, MPa\ (18\, ksi)$

Coefficients C_i	$83 \le S_a \le 359\left(MPa\right)$ $12 \le S_a \le 52\ \left(ksi\right)$	$359 < S_a \le 1793\left(MPa\right)$ $52 < S_a \le 260\ \left(ksi\right)$
1	5.854767E+00	4.940552E+00
2	-1.395072E-01	1.373308E-02
3	-9.597118E-01	-1.385148E-02
4	4.028700E-03	-6.080708E-05
5	4.377509E-02	-1.300476E-05
6	2.487537E-05	0.0
7	-6.795812E-04	0.0
8	-1.517491E-06	0.0
9	1.812235E-06	0.0
10	0.0	0.0
11	0.0	0.0
Note: $E_{FC} = 138E3\ MPa\ (20E3\ ksi)$		

Table 3.F.6 – Coefficients for Fatigue Curve 110.3 – Wrought 70 Copper-Nickel For Temperatures Not Exceeding 371 °C (700 °F) – $\sigma_{ys} = 207\, MPa\ (30\, ksi)$

Coefficients C_i	$62 \le S_a \le 1793\,(MPa)$ $9 \le S_a \le 260\ (ksi)$
1	1.614520E+01
2	7.084155E-02
3	-3.281777E-03
4	3.171113E-02
5	3.768141E-02
6	-1.244577E-03
7	5.462508E-03
8	1.266873E-04
9	2.317630E-04
10	1.346118E-07
11	-3.703613E-07
Notes: $E_{FC} = 138E3\ MPa\ (20E3\ ksi)$	

Table 3.F.7 – Coefficients for Fatigue Curve 110.3 – Wrought 70 Copper-Nickel For Temperatures Not Exceeding 371 °C (700 °F) – $\sigma_{ys} = 310\,MPa\ (45\,ksi)$

Coefficients C_i	$34 \le S_a \le 317 (MPa)$ $4 \le S_a \le 46 (ksi)$	$317 < S_a \le 1793 (MPa)$ $46 < S_a \le 260\ (ksi)$
1	-5.420667E+03	1.016333E+01
2	-3.931295E+02	5.328436E-02
3	-4.778662E+01	-6.492899E-02
4	7.981353E+01	-6.685888E-05
5	2.536083E+02	2.120657E-03
6	1.002901E+00	7.140325E-06
7	2.014578E+00	0.0
8	0.0	0.0
9	0.0	0.0
10	0.0	0.0
11	0.0	0.0

Note: $E_{FC} = 138E3\ MPa\ (20E3\ ksi)$

Table 3.F.8 – Coefficients for Fatigue Curve 110.4 – Nickel-Chromium-Molybdenum-Iron, Alloys X, G, C-4, And C-276 For Temperatures Not Exceeding 427°C (800°F)

Coefficients C_i	$103 \le S_a \le 248 (MPa)$ $15 \le S_a \le 36 (ksi)$	$248 < S_a \le 4881 (MPa)$ $36 < S_a \le 708 (ksi)$
1	5.562508E+00	1.554581E+01
2	-1.014634E+01	6.229821E-02
3	-5.738073E+01	-8.425030E-02
4	7.152267E-01	-8.596020E-04
5	4.578432E+00	1.029439E-04
6	3.584816E-03	8.030748E-06
7	0.0	1.603119E-05
8	0.0	5.051589E-09
9	0.0	-7.849028E-09
10	0.0	0.0
11	0.0	0.0
Note: $E_{FC} = 195E3\ MPa\ (28.3E3\ ksi)$		

Table 3.F.9 – Coefficients for Fatigue Curve 120.1 – High Strength Bolting For Temperatures Not Exceeding 371°C (700°F)

Coefficients C_i	$Maximum\ Nominal\ Stress \leq 2.7S_M$	$Maximum\ Nominal\ Stress \leq 3.0S_M$
	$93 \leq S_a \leq 7929(MPa)$ $13.5 \leq S_a \leq 1150(ksi)$	$37 \leq S_a \leq 7929(MPa)$ $5.3 \leq S_a \leq 1150(ksi)$
1	1.083880E-02	1.268660E+01
2	-4.345648E-01	1.906961E-01
3	1.108321E-01	-8.948723E-03
4	6.215019E-02	-6.900662E-02
5	2.299388E-01	1.323214E-01
6	4.484842E-04	5.334778E-02
7	9.653374E-04	2.322671E-01
8	7.056830E-07	9.260755E-04
9	1.365681E-07	2.139043E-03
10	0.0	1.171078E-06
11	0.0	0.0
Note: $E_{FC} = 206E3\ MPa\ (30E3\ ksi)$		

Table 3.F.10 – Data for Fatigue Curves in Table F.1 Through F.9

Number of Cycles	Fatigue Curve Table					
	3.F.1	3.F.2	3.F.3	3.F.4 Curve A	3.F.4 Curve B	3.F.4 Curve C
1E1	580	420	708	---	---	---
2E1	410	320	512	---	---	---
5E1	275	230	345	---	---	---
1E2	205	175	261	---	---	---
2E2	155	135	201	---	---	---
5E2	105	100	148	---	---	---
8.5E2 (1)	---	---	---	---	---	---
1E3	83	78	119	---	---	---
2E3	64	62	97	---	---	---
5E3	48	49	76	---	---	---
1E4	38	38	64	---	---	---
1.2E4 (1)	---	43	---	---	---	---
2E4	31	36	55.5	---	---	---
5E4	23	29	46.3	---	---	---
1E5	20	26	40.8	---	---	---
2E5	16.5	24	35.9	---	---	---
5E5	13.5	22	31.0	---	---	---
1E6	12.5	20	28.3	28.2	28.2	28.2
2E6	---	---	---	26.9	22.8	22.8
5E6	---	---	---	25.7	19.8	18.4
1E7	11.1	17.8	---	25.1	18.5	16.4
2E7	---	---	---	24.7	17.7	15.2
5E7	---	---	---	24.3	17.2	14.3
1E8	9.9	15.9	---	24.1	17.0	14.1
1E9	8.8	14.2	---	23.9	16.8	13.9
1E10	7.9	12.6	---	23.8	16.6	13.7
1E11	7.0	11.2	---	23.7	16.5	13.6

Table 3.F.10 – Data for Fatigue Curves in Table F.1 Through F.9

Number of Cycles	Fatigue Curve Table					
	3.F.5	3.F.6	3.F.7	3.F.8	3.F.9 (2)	3.F.9 (3)
1E1	260	260	260	708	1150	1150
2E1	190	190	190	512	760	760
5E1	125	125	125	345	450	450
1E2	95	95	95	261	320	300
2E2	73	73	73	201	225	205
5E2	52	52	52	148	143	122
8.5E2 (1)	---	---	46	---	---	---
1E3	44	44	39	119	100	81
2E3	36	36	24.5	97	71	55
5E3	28.5	28.5	15.5	76	45	33
1E4	24.5	24.5	12	64	34	22.5
1.2E4 (1)	---	---	---	---	---	---
2E4	21	19.5	9.6	56	27	15
5E4	17	15	7.7	46.3	22	10.5
1E5	15	13	6.7	40.8	19	8.4
2E5	13.5	11.5	6	35.9	17	7.1
5E5	12.5	9.5	5.2	26.0	15	6
1E6	12.0	9.0	5	20.7	13.5	5.3
2E6	---	---	---	18.7	---	---
5E6	---	---	---	17.0	---	---
1E7	---	---	---	16.2	---	---
2E7	---	---	---	15.7	---	---
5E7	---	---	---	15.3	---	---
1E8	---	---	---	15	---	---
1E9	---	---	---	---	---	---
1E10	---	---	---	---	---	---
1E11	---	---	---	---	---	---

Notes:
1. These data are included to provide accurate representation of the fatigue curves at branches or cusps
2. Maximum Nominal Stress (MNS) less than or equal to 2.7Sm
3. Maximum Nominal Stress (MNS) less than or equal to 3Sm

Table 3.F.11 – Coefficients for the Welded Joint Fatigue Curves

Statistical Basis	Ferritic and Stainless Steels		Aluminum	
	C	h	C	h
Mean Curve	1408.7	0.31950	247.04	0.27712
Upper 68% Prediction Interval $(+1\sigma)$	1688.3	0.31950	303.45	0.27712
Lower 68% Prediction Interval (-1σ)	1175.4	0.31950	201.12	0.27712
Upper 95% Prediction Interval $(+2\sigma)$	2023.4	0.31950	372.73	0.27712
Lower 95% Prediction Interval (-2σ)	980.8	0.31950	163.73	0.27712
Upper 99% Prediction Interval $(+3\sigma)$	2424.9	0.31950	457.84	0.27712
Lower 99% Prediction Interval (-3σ)	818.3	0.31950	133.29	0.27712

Note: In US Customary Units, the equivalent structural stress range parameter, $\Delta S_{ess,k}$, in paragraph 3.F.2.2 and the structural stress effective thickness, t_{ess}, defined in paragraph 5.5.5 are in $ksi/(inches)^{(2-m_{ss})/2m_{ss}}$ and $inches$, respectively. The parameter m_{ss} is defined in paragraph 5.5.5.

Table 3.F.11M – Coefficients for the Welded Joint Fatigue Curves

Statistical Basis	Ferritic and Stainless Steels		Aluminum	
	C	h	C	h
Mean Curve	19930.2	0.31950	3495.13	0.27712
Upper 68% Prediction Interval $(+1\sigma)$	23885.8	0.31950	4293.19	0.27712
Lower 68% Prediction Interval (-1σ)	16629.7	0.31950	2845.42	0.27712
Upper 95% Prediction Interval $(+2\sigma)$	28626.5	0.31950	5273.48	0.27712
Lower 95% Prediction Interval (-2σ)	13875.7	0.31950	2316.48	0.27712
Upper 99% Prediction Interval $(+3\sigma)$	34308.1	0.31950	6477.60	0.27712
Lower 99% Prediction Interval (-3σ)	11577.9	0.31950	1885.87	0.27712

Note: In SI Units, the equivalent structural stress range parameter, $\Delta S_{ess,k}$, in paragraph 3.F.2.2 and the structural stress effective thickness, t_{ess}, defined in paragraph 5.5.5 are in $MPa/(mm)^{(2-m_{ss})/2m_{ss}}$ and mm, respectively. The parameter m_{ss} is defined in paragraph 5.5.5.

PART 4

DESIGN BY RULE REQUIREMENTS

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4.1 General Requirements

4.1.1 Scope

4.1.1.1 The basic requirements for application of the design-by-rules methods of this Division are described in paragraph 4.1. The requirements of Part 4 provide design rules for commonly used pressure vessel shapes under pressure loading and, within specified limits, rules or guidance for treatment of other loadings.

4.1.1.2 Part 4 does not provide rules to cover all loadings, geometries, and details. When design rules are not provided for a vessel or vessel part, a stress analysis in accordance with Part 5 shall be performed considering all of the loadings specified in the User's Design Specification. The user is responsible for defining all applicable loads and conditions acting on the pressure vessel that affect its design. These loads and conditions shall be given in the User's Design Specification. The Manufacturer is not responsible to include any loadings or conditions in the design that are not defined in the User's Design Specification.

4.1.1.3 The design procedures in Part 4 may be used if the allowable stress at the design temperature is governed by time-independent or time-dependent properties unless otherwise noted in a specific design procedure.

4.1.1.4 A screening criterion shall be applied to all vessel parts designed in accordance with this Division to determine if a fatigue analysis is required. The fatigue screening criterion shall be performed in accordance with paragraph 5.5.2. If the results of this screening indicate that a fatigue analysis is required, then the analysis shall be performed in accordance with paragraph 5.5.2. If the allowable stress at the design temperature is governed by time-dependent properties, then a fatigue screening analysis based on experience with comparable equipment shall be satisfied (see paragraph 5.5.2.2).

4.1.1.5 A design-by-analysis in accordance with Part 5 may be used to establish the design thickness and/or configuration (i.e. nozzle reinforcement configuration) in lieu of the design-by-rules in Part 4 for any geometry or loading conditions, see paragraph 4.1.5.1.

4.1.2 Minimum Thickness Requirements

Except for the special provisions listed below, the minimum thickness permitted for shells and heads, after forming and regardless of product form and material, shall be 1.6 mm (0.0625 in.) exclusive of any corrosion allowance. Exceptions are:

a) This minimum thickness does not apply to heat transfer plates of plate-type heat exchangers.

b) This minimum thickness does not apply to the inner pipe of double pipe heat exchangers nor to pipes and tubes that are enclosed and protected by a shell, casing or ducting, where such pipes or tubes are DN 150 (NPS 6) and less. This exemption applies whether or not the outer pipe or shell is constructed to Code rules. All other pressure parts of these heat exchangers that are constructed to Code rules must meet the 1.6 mm (0.0625 in.) minimum thickness requirements.

c) The minimum thickness of shells and heads used in compressed air service, steam service, and water service, made from carbon or low alloy steel materials shall be 2.4 mm (0.0938 in.) exclusive of any corrosion allowance.

d) This minimum thickness does not apply to the tubes in air cooled and cooling tower heat exchangers if all of the following provisions are met:

    1) The tubes shall be protected by fins or other mechanical means.

    2) The tube outside diameter shall be a minimum of 10 mm (0.375 in.) and a maximum of 38 mm (1.5 in.).

3) The minimum thickness used shall not be less than that calculated by the equations given in paragraph 4.3 and in no case less than 0.5 mm (0.022 in.).

4.1.3 Material Thickness Requirements

4.1.3.1 Allowance For Fabrication – The selected thickness of material shall be such that the forming, heat treatment, and other fabrication processes will not reduce the thickness of the material at any point below the minimum required thickness.

4.1.3.2 Mill Undertolerance – Plate material shall be ordered not thinner than the minimum required thickness. Vessels made of plate furnished with an undertolerance of not more than the smaller value of 0.3 mm (0.01 in.) or 6% of the ordered thickness may be used at the full maximum allowable working pressure for the thickness ordered. If the specification to which the plate is ordered allows a greater undertolerance, the ordered thickness of the materials shall be sufficiently greater than the design thickness so that the thickness of the material furnished is not more than the smaller of 0.3 mm (0.01 in.) or 6% under the design thickness.

4.1.3.3 Pipe Undertolerance – If pipe or tube is ordered by its nominal wall thickness, the manufacturing undertolerance on wall thickness shall be taken into account. After the minimum wall thickness is determined, it shall be increased by an amount sufficient to provide the manufacturing undertolerance allowed in the pipe or tube specification.

4.1.4 Corrosion Allowance in Design Equations

4.1.4.1 The dimensional symbols used in all design equations and figures throughout this Division represent dimensions in the corroded condition.

4.1.4.2 The term corrosion allowance as used in this Division is representative of loss of metal by corrosion, erosion, mechanical abrasion, or other environmental effects and shall be accounted for in the design of vessels or parts when specified in the User's Design Specification.

4.1.4.3 The user shall determine the required corrosion allowance over the life of the vessel and specify such in the User's Design Specification. The Manufacturer shall add the required allowance to all minimum required thicknesses in order to arrive at the minimum ordered material thickness. The corrosion allowance need not be the same for all parts of a vessel. If corrosion or other means of metal loss do not exist, then the user shall specify in the User's Design Specification that a corrosion allowance is not required.

4.1.5 Design Basis

4.1.5.1 Design Thickness – The design thickness of the vessel part shall be determined using the design-by-rule methods of Part 4 with the load and load case combinations specified in paragraph 4.1.5.3. Alternatively, the design thickness may be established using the design-by-analysis procedures in Part 5, even if this thickness is less than that established using Part 4 design-by-rule methods. In either case, the design thickness shall not be less than the minimum thickness specified in paragraph 4.1.2.

4.1.5.2 Definitions – the following definitions shall be used to establish the design basis of the vessel. Each of these parameters shall be specified in the User's Design Specification.

a) Design Pressure – The pressure used in the design of a vessel component together with the coincident design metal temperature, for the purpose of determining the minimum permissible thickness or physical characteristics of the different zones of the vessel. Where applicable, static head and other static or dynamic loads shall be included in addition to the specified design pressure [2.2.2.1.d.1] in the determination of the minimum permissible thickness or physical characteristics of a particular zone of the vessel.

b) Maximum Allowable Working Pressure – The maximum gage pressure permissible at the top of a completed vessel in its normal operating position at the designated coincident temperature for that pressure. This pressure is the least of the values for the internal or external pressure to be determined by the rules of this Division for any of the pressure boundary parts, considering static head thereon, using nominal thicknesses exclusive of allowances for corrosion and considering the effects of any combination of loadings specified in the User's Design Specification at the designated coincident temperature. It is the basis for the pressure setting of the pressure relieving devices protecting the vessel. The specified design pressure may be used in all cases in which calculations are not made to determine the value of the maximum allowable working pressure.

c) Test Pressure – The test pressure is the pressure to be applied at the top of the vessel during the test. This pressure plus any pressure due to static head at any point under consideration is used in the applicable design equations to check the vessel under test conditions.

d) Design Temperature and Coincident Pressure – The design temperature for any component shall not be less than the mean metal temperature expected coincidentally with the corresponding maximum pressure (internal and, if specified, external). If necessary, the mean metal temperature shall be determined by computations using accepted heat transfer procedures or by measurements from equipment in service under equivalent operating conditions. In no case shall the metal temperature anywhere within the wall thickness exceed the maximum temperature limit in paragraph 4.1.5.2.d.1.

   1) A design temperature greater than the maximum temperature listed for a material specification in Annex 3.A is not permitted. In addition, if the design includes external pressure (see paragraph 4.4), then the design temperature shall not exceed the temperature limits specified in Table 4.4.1.

   2) The maximum design temperature marked on the nameplate shall not be less than the expected mean metal temperature at the corresponding MAWP.

   3) When the occurrence of different mean metal temperatures and coincident pressures during operation can be accurately predicted for different zones of a vessel, the design temperature for each of these zones may be based on the predicted temperatures. These additional design metal temperatures with their corresponding MAWP, may be marked on the nameplate as required.

e) Minimum Design Metal Temperature and Coincident Pressure – The minimum design metal temperature (MDMT) shall be the coldest expected in normal service, except when colder temperatures are permitted by paragraph 3.11. The MDMT shall be determined by the principles described in paragraph 4.1.5.2.d. Considerations shall include the coldest operating temperature, operational upsets, auto refrigeration, atmospheric temperature, and any source of cooling.

   1) The MDMT marked on the nameplate shall correspond to a coincident pressure equal to the MAWP.

   2) When there are multiple MAWP, the largest value shall be used to establish the corresponding MDMT marked on the nameplate.

   3) When the occurrence of different MDMT and coincident pressures during operation can be accurately predicted for different zones of a vessel, the MDMT for each of these zones may be based on the predicted temperatures. These additional MDMT together with their corresponding MAWP, may be marked on the nameplate as required.

4.1.5.3 Design Loads And Load Case Combinations – All applicable loads and load case combinations shall be considered in the design to determine the minimum required wall thickness for a vessel part.

a) The loads that shall be considered in the design shall include, but not be limited to, those shown in Table 4.1.1 and shall be included in the User's Design Specification.

b) The load combinations that shall be considered shall include, but not be limited to, those shown in Table 4.1.2.

c) When analyzing a loading combination, the value of allowable stress shall be evaluated at the coincident temperature.

d) Combinations of loads that result in a maximum thickness shall be evaluated. In evaluating load cases involving the pressure term, P, the effects of the pressure being equal to zero shall be considered. For example, the maximum difference in pressure that may exist between the inside and outside of the vessel at any point or between two chambers of a combination unit or, the conditions of wind loading with an empty vertical vessel at zero pressure may govern the design.

e) The applicable loads and load case combinations shall be specified in the User's Design Specification.

f) If the vessel or part is subject to cyclic operation and a fatigue analysis is required (see paragraph 4.1.1.4), then a pressure cycle histogram and corresponding thermal cycle histogram shall be provided in the User's Design Specification.

4.1.6 Design Allowable Stress

4.1.6.1 Design Condition – The allowable stresses for all permissible materials of construction are provided in Annex 3.A. The wall thickness of a vessel computed by the rules of Part 4 for any combination of loads (see paragraph 4.1.5) that induce primary stress (see paragraph 5.12.17) and are expected to occur simultaneously during operation shall satisfy the equations shown below.

$$P_m \le S \tag{4.1.1}$$

$$P_m + P_b \le 1.5S \tag{4.1.2}$$

4.1.6.2 Test Condition – The allowable stress for the test condition shall be established by the following requirements. Controls shall be provided to ensure that the Test Pressure is limited such that these allowable stresses are not exceeded. When applicable, the static head and any additional pressure loadings shall be included.

a) Hydrostatically Tested Vessels – when a hydrostatic test is performed in accordance with Part 8, the hydrostatic test pressure of a completed vessel shall not exceed that value which results in the following equivalent stress limits:

1) A calculated P_m shall not exceed the applicable limit given below:

$$P_m \le 0.95S_y \tag{4.1.3}$$

2) A calculated $P_m + P_b$ shall not exceed the applicable limits given below:

$$P_m + P_b \le 1.43S_y \qquad \text{for } P_m \le 0.67S_y \tag{4.1.4}$$

$$P_m + P_b \le \left(2.43S_y - 1.5P_m\right) \qquad \text{for } 0.67S_y < P_m \le 0.95S_y \tag{4.1.5}$$

b) Pneumatically Tested Vessels – when a pneumatic test is performed in accordance with Part 8, the pneumatic test pressure of a completed vessel shall not exceed that value which results in the following equivalent stress limits:

1) A calculated P_m shall not exceed the applicable limit given below:

$$P_m \le 0.80S_y \tag{4.1.6}$$

2) A calculated $P_m + P_b$ shall not exceed the applicable limits given in below:

$$P_m + P_b \le 1.20 S_y \qquad for\ P_m \le 0.67 S_y \tag{4.1.7}$$

$$P_m + P_b \le \left(2.20 S_y - 1.5 P_m\right) \qquad for\ \ 0.67 S_y < P_m \le 0.8 S_y \tag{4.1.8}$$

4.1.7 Materials in Combination

Except as specifically prohibited by the rules of this Division, a vessel may be designed for and constructed of any combination of materials listed in Part 3. For vessels operating at temperatures other than ambient temperature, the effects of differences in coefficients of thermal expansion of dissimilar materials shall be considered.

4.1.8 Combination Units

4.1.8.1 Combination Unit – is a pressure vessel that consists of more than one independent pressure chamber, operating at the same or different pressures and temperatures. The parts separating each independent pressure chamber are the common elements. Each element, including the common elements, shall be designed for at least the most severe condition of coincident pressure and temperature expected in normal operation. Only the parts of chambers that come within the scope of this Division need be constructed in compliance with its provisions. Additional design requirements for chambers classified as jacketed vessels are provided in paragraph 4.11.

4.1.8.2 Common Element Design – It is permitted to design each common element for a differential pressure less than the maximum of the design pressures of its adjacent chambers (differential pressure design) or a mean metal temperature less than the maximum of the design temperatures of its adjacent chambers (mean metal temperature design), or both, only when the vessel is to be installed in a system that controls the common element operating conditions.

a) Differential Pressure Design – When differential pressure design is permitted, the common element design pressure shall be the maximum differential design pressure expected between the adjacent chambers. The common element and its corresponding differential pressure shall be indicated in the "Remarks" section of the Manufacturer's Data Report (see paragraph 2.3.4) and marked on the vessel (see Annex 2.F). The differential pressure shall be controlled to ensure the common element design pressure is not exceeded.

b) Mean Metal Temperature Design – When mean metal temperature design is used, the maximum common element design temperature determined in accordance with 4.1.5.2.d may be less than the greater of the maximum design temperature of its adjacent chambers; however, it shall not be less than the lower of the maximum design temperatures of its adjacent chambers. The common element and its corresponding design temperature shall be indicated in the "Remarks" section of the Manufacturer's Data Report (see paragraph 2.3.4) and marked on the vessel (see Annex 2.F). The fluid temperature, flow and pressure, as required, shall be controlled to ensure the common element design temperature is not exceeded.

4.1.9 Cladding and Weld Overlay

4.1.9.1 The design calculations for integrally clad plate or overlay weld clad plate may be based on a thickness equal to the nominal thickness of the base plate plus S_C / S_B times the nominal thickness of the cladding, less any allowance provided for corrosion, provided all of the following conditions are met.

a) The clad plate conforms to one of the specifications listed in the tables in Part 3 or is overlay weld clad plate conforming to Part 3.

b) The joints are completed by depositing corrosion resisting weld metal over the weld in the base plate to restore the cladding.

c) The allowable stress of the weaker material is at least 70% of the allowable stress of the stronger material.

4.1.9.2 When S_C is greater than S_B, the multiplier S_C/S_B shall be taken equal to unity.

4.1.10 Internal Linings

Corrosion resistant or abrasion resistant linings are those not integrally attached to the vessel wall, i.e., they are intermittently attached or not attached at all. In either case, such linings shall not be given any credit when calculating the thickness of the vessel wall.

4.1.11 Flanges and Pipe Fittings

4.1.11.1 The following standards covering flanges and pipe fittings are acceptable for use under this Division in accordance with the requirements of Part 1.

a) ASME B16.5, Pipe Flanges and Flanged Fittings

b) ASME B16.9, Factory-Made Wrought Steel Butt-welding Fittings

c) ASME B16.11, Forged Fittings, Socket- Welding and Threaded

d) ASME B16.15, Cast Bronze Threaded Fittings, Classes 125 and 250

e) ASME B16.20, Metallic Gaskets for Pipe Flanges – Ring-Joint, Spiral-Wound, and Jacketed

f) ASME B16.24, Cast Copper Alloy Pipe Flanges and Flanged Fittings, Class 150, 300, 400, 600, 900, 1500, and 2500

g) ASME B16.47, Large Diameter Steel Flanges, NPS 26 Through NPS 60

4.1.11.2 Pressure-temperature ratings shall be in accordance with the applicable standard except that the pressure-temperature ratings for ASME B16.9 and ASME B16.11 fittings shall be calculated as for straight seamless pipe in accordance with the rules of this Division including the maximum allowable stress for the material.

4.1.11.3 A forged nozzle flange (i.e. long weld neck flange) may be designed using the ASME B16.5/B16.47 pressure-temperature ratings for the flange material being used, provided all of the following are met.

a) For ASME B16.5 applications, the forged nozzle flange shall meet all dimensional requirements of a flanged fitting given in ASME B16.5 with the exception of the inside diameter. The inside diameter of the forged nozzle flange shall not exceed the inside diameter of the same size and class lap joint flange given in ASME B16.5. For ASME B16.47 applications, the inside diameter shall not exceed the weld hub diameter "A" given in the ASME B16.47 tables.

b) For ASME B16.5 applications, the outside diameter of the forged nozzle neck shall be at least equal to the hub diameter of the same size and class ASME B16.5 lap joint flange. For ASME B16.47 applications, the outside diameter of the hub shall at least equal the "X" diameter given in the ASME B16.47 tables. Larger hub diameters shall be limited to nut stop diameter dimensions (see paragraph 4.16).

4.1.12 Nomenclature

P_m general primary membrane stress (see Part 5).

$P_m + P_b$ general primary membrane plus primary bending stress (see Part 5).

S_B allowable stress from Annex 3.A at the design temperature for the base plate at the design temperature.

S_C allowable stress from Annex 3.A at the design temperature for the cladding or, for the weld overlay, the allowable stress of the wrought material whose chemistry most closely approximates that of the cladding at the design temperature.

S allowable stress from Annex 3.A at the design temperature.

S_y yield strength at the test temperature evaluated in accordance with Annex 3.D.

4.1.13 Tables

Table 4.1.1 – Design Loads

Design Load Parameter	Description
P	Internal or External Specified Design Pressure (see paragraph 4.1.5.2.a)
P_s	Static head from liquid or bulk materials (e.g. catalyst)
D	Dead weight of the vessel, contents, and appurtenances at the location of interest, including the following: • Weight of vessel including internals, supports (e.g. skirts, lugs, saddles, and legs), and appurtenances (e.g. platforms, ladders, etc.) • Weight of vessel contents under operating and test conditions • Refractory linings, insulation • Static reactions from the weight of attached equipment, such as motors, machinery, other vessels, and piping
L	• Appurtenance Live loading • Effects of fluid flow, steady state or transient • Loads resulting from wave action
E	Earthquake loads (see ASCE 7 for the specific definition of the earthquake load, as applicable)
W	Wind Loads
S	Snow Loads
F	Loads due to Deflagration

Table 4.1.2 – Design Load Combinations

Design Load Combination (1)	General Primary Membrane Allowable Stress (2)
$P+P_s+D$	S
$P+P_s+D+L$	S
$P+P_s+D+S$	S
$0.9P+P_s+D+0.75L+0.75S$	S
$0.9P+P_s+D+(W\ or\ 0.7E)$	S
$0.9P+P_S+D+0.75(W\ or\ 0.7E)+0.75L+0.75S$	S
$0.6D+(W\ or\ 0.7E)$ (3)	S
P_s+D+F	See Annex 4.D

Notes
1) The parameters used in the Design Load Combination column are defined in Table 4.1.1.
2) S is the allowable stress for the load case combination (see paragraph 4.1.5.3.c)
3) This load combination addresses an overturning condition. If anchorage is included in the design, consideration of this load combination is not required.

4.2 Design Rules for Welded Joints

4.2.1 Scope

Design requirements for weld joints are provided in paragraph 4.2. Acceptable weld joint details are provided for most common configurations. Alternative details may be used if they can be qualified by a design procedure using Part 5. Rules for sizing welds are also provided.

4.2.2 Weld Category

The term weld category defines the location of a joint in a vessel, but not the weld joint type. The weld categories established by this paragraph are for use elsewhere in this Division in specifying special requirements regarding joint type and degree of examination for certain welded pressure joints. Since these special requirements that are based on thickness do not apply to every welded joint, only those joints to which special requirements apply are included in categories. The weld categories are defined in Table 4.2.1 and shown in Figure 4.2.1.

4.2.3 Weld Joint Type

The weld joint type defines the type of weld between pressure and/or nonpressure parts. The definitions for the weld joint types are shown in Table 4.2.2.

4.2.4 Weld Joint Efficiency

The weld joint efficiency of a welded joint is expressed as a numerical quantity and is used in the design of a joint as a multiplier of the appropriate allowable stress value taken from Annex 3.A. The weld joint efficiency shall be determined from Table 7.2.

4.2.5 Types of Joints Permitted

4.2.5.1 Definitions

a) Butt Joint – A butt joint is a connection between the edges of two members with a full penetration weld. The weld is a double sided or single sided groove weld that extends completely through both of the parts being joined.

b) Corner Joint – A corner joint is a connection between two members at right angles to each other in the form of an L or T that is made with a full or partial penetration weld, or fillet welds. Welds in full penetration corner joints shall be groove welds extending completely through at least one of the parts being joined and shall be completely fused to each part.

c) Angle Joint – An angle joint is a connection between the edges of two members with a full penetration weld with one of the members consisting of a transition of diameter. The weld is a double sided or single sided groove weld that extends completely through both of the parts being joined.

d) Spiral Weld - a weld joint having a helical seam.

e) Fillet Weld – A fillet weld is a weld that is approximately triangular in cross section that joins two surfaces at approximately right angles to each other.

f) Gross Structural Discontinuity – A gross structural discontinuity is a source of stress or strain intensification which affects a relatively large portion of a structure and has a significant effect on the overall stress or strain pattern or on the structure as a whole. Examples of gross structural discontinuities are head-to-shell and flange-to-shell junctions, nozzles, and junctions between shells of different diameters or thicknesses.

g) Lightly Loaded Attachments – Weld stress due to mechanical loads on attached member not over 25% of allowable stress for fillet welds and temperature difference between shell and attached member not expected to exceed 14°C (25°F) shall be considered lightly loaded.

h) Minor Attachments – Parts of small size, 10 mm (0.375 in.) thick or 82 cm^3 (5 $in.^3$) in volume, that carry no load or an insignificant load such that a stress calculation in designer's judgment is not required; examples include nameplates, insulation supports, and locating lugs.

i) Major Attachments –Parts that are not minor or lightly loaded as described above.

4.2.5.2 Category A Locations

a) All joints of Category A shall be Type No. 1 butt joints.

b) Acceptable Category A welds are shown in Tables 4.2.4 and 4.2.5.

c) Transition Joints Between Sections Of Unequal Thickness – Unless the requirements of Part 5 are shown to be satisfied, a tapered transition shall be provided at joints between sections that differ in thickness by more than one-fourth of the thickness of the thinner section or by more than 3 mm (0.125 in.). The transition may be formed by any process that will provide a uniform taper. When Part 5 is not used, the following additional requirements shall also apply.

   1) When a taper is required on any shell section intended for butt welded attachment, the transition geometry shall be in accordance with Table 4.2.4, Details 4, 5, and 6.

   2) When a taper is required on a hemispherical head intended for butt welded attachment, the transition geometry shall be in accordance with Table 4.2.5, Details 2, 3, 4 and 5.

   3) A hemispherical head which has a greater thickness than a cylinder of the same inside diameter may be machined to the outside diameter of the cylinder provided the remaining thickness is at least as great as that required for a shell of the same diameter.

   4) When the transition is formed by adding additional weld metal beyond what would otherwise be the edge of the weld, such additional weld metal buildup shall be subject to the requirements of Part 6. The butt weld may be partly or entirely in the tapered section.

   5) The requirements of this paragraph do not apply to flange hubs.

4.2.5.3 Category B Locations

a) The joints of Category B may be any of the following types:

   1) Type No. 1 butt joints,

   2) Type No.2 butt joints except as limited in paragraph 4.2.5.7,

   3) Type No. 3 butt joints may only be used for shells having a thickness of 16 mm (0.625 in.) or less and a diameter of 610 mm (or 24 in.) and less.

b) Acceptable Category B welds are shown in Tables 4.2.4 and 4.2.5.

c) Backing strips shall be removed from Type No. 2 butt joints unless access conditions prevent their removal. If a fatigue analysis of Type No. 2 butt joints with a backing strip in place is required, then a stress concentration factor of 2.0 for membrane stresses and of 2.5 for bending stress shall be applied.

d) Transition joints between shell sections of unequal thickness shall meet the requirements of paragraph 4.2.5.2.c and shall be in accordance with Table 4.2.4 and Table 4.2.5. An ellipsoidal head which has a greater thickness than a cylinder of the same inside diameter may be machined to the outside diameter of the cylinder provided the remaining thickness is at least as great as that required for a shell of the same diameter.

e) Transition joints between nozzle necks and attached piping of unequal thickness shall be made using a tapered transition in accordance with Table 4.2.4, Details 7 and 8.

f) When butt joints are required elsewhere in this Division for Category B, an angle joint connecting a transition in diameter to a cylinder shall be considered as meeting this requirement provided the requirements of Type No. 1 butt joints are met. All requirements pertaining to the butt joint shall apply to the angle joint.

4.2.5.4 Category C Locations

a) The joints of Category C may be any of the following types:

   1) Type No. 1 butt joints,

   2) Full penetration corner joints except as limited in paragraph 4.2.5.7.

3) Fillet welded joints for the attachment of loose type flanges shown in Table 4.2.9, with the following limitations:

   i) The materials of the flange and the part it is welded to are Type 1 Materials (see Table 4.2.3).

   ii) The minimum specified yield strength of both materials is less than 552MPa (80 ksi).

   iii) The minimum elongation of both materials is 12% in 50 mm (2 in.) gauge length.

   iv) The thickness of the materials to which the flange is welded does not exceed 32 mm (1.25 in.).

   v) The fillet weld dimensions satisfy the requirements shown in Table 4.2.9.

   vi) A fatigue-screening criterion shall be performed in accordance with paragraph 5.5.2 to determine if a fatigue analysis is required. If the results of this screening indicate that a fatigue analysis is required, then the analysis shall be performed in accordance with paragraph 5.5.2.

   vii) Loose type flanges that do not conform to ASME B16.5 are only permitted when both of the following requirements are satisfied.

   - The material of construction for the flange satisfies the following equation.

$$\frac{S_{yT}}{S_u} \leq 0.625 \tag{4.2.1}$$

   - The component is not in cyclic service, i.e. a fatigue analysis is not required in accordance with paragraph 4.1.1.4.

b) Acceptable Category C welds are shown in the following tables.

   1) Table 4.2.4 – Some acceptable weld joints for shell seams.

   2) Table 4.2.6 – Some acceptable weld joints for unstayed flat heads, tubesheets without a bolting flange, and side plates of rectangular pressure vessels

   3) Table 4.2.7 – Some acceptable weld joints with butt weld hubs.

   4) Table 4.2.8 – Some acceptable weld joints for attachment of tubesheets with a bolting flange

   5) Table 4.2.9 – Some acceptable weld joints for flange attachments.

c) Flat Heads, Lap Joint Stub Ends, and Tubesheets with Hubs for Butt Joints

   1) Hubs for butt welding to the adjacent shell, head, or other pressure parts such as tubesheets and flat heads as shown in Table 4.2.7 shall be forged or machined from flat plate. Forged hubs shall be forged in such a manner as to provide in the hub the full minimum tensile strength and elongation specified for the material in the direction parallel to the axis of the vessel. Proof of this shall be furnished by a tension test specimen (subsize, if necessary) taken in this direction and as close to the hub as practical. Hubs machined from flat plates should satisfy the requirements of paragraph 3.9.

   2) Flanges with hubs as shown in Table 4.2.9, Details 6, 7, and 8 shall not be machined from plate.

d) Corner Joints – If shells, heads, or other pressure parts are welded to a forged or rolled plate to form a corner joint as shown in Table 4.2.6 and Table 4.2.8, then the welds shall meet the following requirements.

   1) On the cross section through the welded joint, the line between the weld metal and the forged or rolled plate being attached shall be projected on planes both parallel to and perpendicular to the surface of the plate being attached, in order to determine the dimensions a and b, respectively.

   2) The dimensional requirements on a and b shall meet the applicable requirements in Tables 4.2.6 and Table 4.2.8.

3) Weld joint details that have a dimension through the joint that is less than the thickness of the shell, head, or other pressure part, or that provide attachment eccentric thereto are not permitted.
4) If an integral tubesheet is located between two shells, heads, or other pressure parts, then a weld attachment detail as shown in Table 4.2.6 shall be used for each attachment.

4.2.5.5 Category D Locations

a) The joints of Category D may be any of the following types.
   1) Type No. 1 butt joints
   2) Full penetration corner joints except as limited in paragraph 4.2.5.7
   3) Full penetration corner joints at the nozzle neck or fillet welds, or both
   4) Partial penetration corner joint at the nozzle neck
b) Acceptable Category D welds are shown in the following tables.
   1) Table 4.2.4 – Some acceptable weld joints for shell seams
   2) Table 4.2.10 – Some acceptable full penetration welded nozzle attachments not readily radiographable
   3) Table 4.2.11 – Some acceptable pad welded nozzle attachments and other connections to shells
   4) Table 4.2.12 – Some acceptable fitting type welded nozzle attachments and other connections to shells
   5) Table 4.2.13 – Some acceptable welded nozzle attachments that are readily radiographable
   6) Table 4.2.14 – Some acceptable partial penetration nozzle attachments
c) Requirements for nozzle welds are shown below.
   1) Type No. 1 butt joints or full penetration joints shall be used when the opening in a shell is 64 mm (2.5 in.) or more in thickness.
   2) Nozzle Neck Abutting The Vessel Wall Without Reinforcement – Nozzle necks abutting the vessel wall without added reinforcing element shall be attached by a full penetration groove weld. Backing strips shall be used with welds deposited from only one side or when complete joint penetration cannot be verified by visual inspection. Backing strips, when used, shall be removed after welding. Permissible types of weld attachments are shown in Table 4.2.10, Details 1, 2, and 8.
   3) Insert Nozzle Necks Without Reinforcement – Nozzle necks without added reinforcing elements inserted partially into or through a hole cut in the vessel wall shall be attached by a full penetration groove weld. Backing strips, when used, shall be removed after welding. Permissible types of weld attachments are shown in Table 4.2.10, Details 3, 4, 5, 6, 7, and 9.
   4) Insert Nozzle Necks With Reinforcement – Inserted type necks having added reinforcement in the form of one or more separate reinforcing plates shall be attached by continuous welds at the outer edge of the reinforcement plate and at the nozzle neck periphery. A fatigue-screening criterion shall be applied to nozzles with separate reinforcement and non-integral attachment designs. The welds attaching the neck to the vessel wall and to the reinforcement shall be full penetration groove welds. Permissible types of weld attachments are shown in Table 4.2.11, Details 1, 2, and 3. (Also see 4.2.5.5.d)
   5) Studded Pad Type Connections – Studded connections that may have externally imposed loads shall be attached using full penetration welds in accordance with Table 4.2.11, Detail 5. Studded pad type connections on which there are essentially no external loads, such as manways and handholes used only as inspection openings, thermowell connections, etc., may be attached using fillet weld in accordance with Table 4.2.11, Detail 4.
   6) Fittings With Internal Threads – Internally threaded fittings shall be limited to NPS 2 or smaller. Permissible types of weld attachments are shown in Table 4.2.12.

7) Nozzles With Integral Reinforcement – Nozzles having integral reinforcement may be attached using butt welds of Type No.1. Nozzles or other connections with integral reinforcement that are attached with corner welds shall be attached by means of full penetration corner welds. Permissible types of weld attachments are shown in Table 4.2.13.
8) Nozzle Attached With Partial Penetration Welds – Partial penetration welds may be used only for nozzle attachments, such as instrumentation openings, inspection openings, etc., on which there are essentially no external mechanical loadings and on which there will be no thermal stresses greater than in the vessel itself. Permissible types of weld attachments are shown in Table 4.2.14. If Table 4.2.14, Details 3 and 4 are used, then the material in the neck shall not be included in the reinforcement area calculation (see paragraph 4.5).

d) Except for nozzles at small ends of cones reinforced in accordance with the requirements of paragraphs 4.3.11, 4.3.12, 4.4.13, and 4.4.14, as applicable, added reinforcement in the form of separate reinforcing plates or pads may be used provided the vessel and nozzles meet all of the following requirements.
1) The materials of the nozzle, pad, and vessel wall conform to those listed in Section IX, Table QW/QB-422 for Material Types 1 and 4 shown in Table 4.2.3.
2) The specified minimum tensile strength of the nozzle, pad, and vessel wall materials does not exceed 550 MPa (80 ksi).
3) The minimum elongation of the nozzle, pad, and vessel wall materials is 12% in 50 mm (2 in.).
4) The thickness of the added reinforcement does not exceed 1.5 times the vessel wall thickness.
5) The requirements of paragraph 5.5 for pads, i.e. non-integral construction, in cyclic service are met.

4.2.5.6 Category E Locations

a) Method Of Attachment – Attachment of nonpressure parts shall be in accordance with the following requirements.
1) Nonpressure parts, supports, lugs, brackets, and stiffeners may be attached to the inside or outside wall using butt welds, full penetration groove welds, partial penetration welds, fillet welds, or stud welds as limited in the subsequent paragraphs.
2) Resistance welded studs may be used for minor attachments to nonpressure parts for all materials except those included in Material Type 2 (see Table 4.2.3).
3) Supports, lugs, brackets, stiffeners, and other attachments may be attached with stud bolts to the outside or inside of a vessel wall (see paragraph 4.15.5).
4) All attachments shall conform to the curvature of the shell to which they are to be attached.
5) All welds joining minor attachments, see paragraph 4.2.5.1.g, to pressure parts may be continuous or non-continuous for Material Types 1, 3, and 4 (see Table 4.2.3).
6) All welds joining nonpressure parts to pressure parts shall be continuous for Material Type 2 (see Table 4.2.3).
7) Some acceptable types of attachment weld and associated minimum weld sizes are shown in Figure 4.2.2, see paragraphs 4.2.5.6.e and 4.2.5.6.f for limitations.
8) Some acceptable methods of attaching stiffening rings are shown in Figure 4.2.3, see paragraphs 4.2.5.6.e and 4.2.5.6.f for limitations.

b) Materials for Major Attachments to Pressure Parts – Attachments welded directly to pressure parts shall be of a material listed in Annex 3.A.
1) The material and the deposited weld metal shall be compatible with that of the pressure part.
2) For Material Type 2 (see Table 4.2.3), all permanent structural attachments that are welded directly to pressure parts shall be made of materials whose specified minimum yield strength is within ±20% of that of the material to which they are attached. An exception to this requirement is that lightly loaded attachments of non-hardenable austenitic stainless steels conforming to either SA-240, SA-312, or SA-479 are permitted to be fillet welded to pressure parts conforming to either SA-353, SA-553 Type 1 and Type 2, or SA-645.

c) Materials for Minor Attachments to Pressure Parts – Except as limited by paragraph 4.2.5.6.b or for forged fabrication (see paragraph 6.7), minor attachments may be of non-certified material and may be welded directly to the pressure part provided the requirements shown below are satisfied.
   1) The material is identified and is suitable for welding.
   2) The material is compatible insofar as welding is concerned with that to which the attachment is to be made.
   3) The welds are postweld heat treated when required in Part 6.
d) Materials for Attachments Welded to Nonpressure Parts – Attachments welded to nonpressure parts may be of non-certified material, provided the material is identified, is suitable for welding, and is compatible with the material to which attachment is made.
e) Attachment Welds To Pressure Parts Of Material Types 1 and 4 (see Table 4.2.3) – Welds attaching nonpressure parts or stiffeners to pressure parts shall be one of the following:
   1) A fillet weld not over 13 mm (0.5 in.) leg dimension and the toe of the weld not closer than $\sqrt{Rt_s}$ from a gross structural discontinuity
   2) A partial penetration weld plus fillet weld; this is limited to the attachment of parts not exceeding 38 mm (1.5 in.) in thickness
   3) A full penetration groove weld plus a fillet weld on each side
   4) A full penetration butt weld; the prior deposition of weld metal to provide a boss for the butt weld is permissible provided it is checked for soundness by suitable nondestructive examination. Heat treatment for the weld build-up region shall be considered.
   5) For attachment of support skirts or other supports involving similar attachment orientation, in addition to the weld types of paragraphs 4.2.5.6.e.3 and 4.2.5.6.e.4, welds of greater effective throat dimension than 90 deg fillet welds, as obtained by increased leg dimension or angle and bevel of parts joined, may be used where the effective throat is t_a (see Figure 4.2.4); however, the limitation on thickness in paragraph 4.2.5.6.e.2 shall apply.
   6) Stiffening rings may be stitch welded when the material of construction satisfies Equation (4.2.1) and the component is not in cyclic service, i.e. a fatigue analysis is not required in accordance with paragraph 4.1.1.4.
f) Attachment Welds To Pressure Parts of Material Types 2 and 3 (see Table 4.2.3) – Welds attaching nonpressure parts or stiffeners to pressure parts shall be one of the following:
   1) Except as permitted in paragraphs 4.2.5.6.f.2, fillet welds are permissible only for seal welds or for lightly loaded attachments with a weld size not over 10 mm (0.375 in.) leg dimension and the toe of the weld shall not be located closer than $\sqrt{Rt_s}$ from a gross structural discontinuity.
   2) For materials SA-333 Grade 8, SA-334 Grade 8, SA-353, SA-522, SA-553, and SA-645, fillet welds are permissible, provided that the fillet weld leg dimension does not exceed 13 mm (0.5 in.) and the toe of the weld shall not be located closer than $\sqrt{Rt_s}$ from another gross structural discontinuity.
   3) A partial penetration weld plus fillet weld; limited to the attachment of parts not exceeding 19 mm (0.75 in.) in thickness.
   4) A full penetration groove weld plus a fillet weld on each side.
   5) Full penetration butt weld (see paragraph 4.2.5.6.e.4 for boss requirements).
   6) For attachment of support skirts or other supports involving similar attachment orientation, in addition to welds permitted by paragraph 4.2.5.6.f.5 above, welds of greater effective throat dimension than 90 deg fillet welds may be used where the throat is a minimum of t_a (see Figure 4.2.4). The details in this figure are limited to attachment of parts not exceeding 19 mm (0.75 in.) in thickness unless the attachment weld is double welded.

g) Stress Values For Weld Material – Attachment weld strength shall be based on the nominal weld area and the allowable stress values in Annex 3.A for the weaker of the two materials joined, multiplied by the reduction factors, W_r, shown below.

   1) Full penetration butt or groove welds – $W_r = 1.0$; the nominal weld area is the depth of the weld times the length of weld.

   2) Partial penetration groove or partial penetration groove plus fillet welds – $W_r = 0.75$; the nominal weld area is:

      i) Groove welds – the depth of penetration times the length of weld.

      ii) Groove welds with fillet welds – the combined throat and depth of penetration, exclusive of reinforcement, times the length of weld.

h) Fillet welds – $W_r = 0.5$; the nominal weld area is the throat area.

i) Weld Overlay And Clad Construction

   1) Attachments may be welded directly to weld overlay deposits without restriction.

   2) For clad construction where design credit is taken is taken for cladding thickness, attachment welds may be made directly to the cladding based for loadings not producing primary stress in the attachment weld not exceeding 10% of the design allowable stress value of the attachment or the cladding material, whichever is less. As an alternative, local regions of weld overlay can be located within the cladding to provide an attachment location.

   3) For applied linings, attachments should be made directly to the base metal unless an analysis, tests, or both can be performed to establish the adequacy and reliability of an attachment made directly to the lining. Note that successful experience with similar linings in comparable service may provide a basis for judgment.

j) PWHT Requirements – For heat treatment after welding, the fabrication requirements of the vessel base metal apply.

k) Evaluation Of Need For Fatigue Analysis – In applying the fatigue screening analysis in paragraph 5.5.2, fillet welds and non-full-penetration welds shall be considered to be nonintegral attachments, except that the following welds need not be considered because of the limitations of their use:

   1) Welds covered by paragraphs 4.2.5.6.c, 4.2.5.6.e.1, 4.2.5.6.f.1 and 4.2.5.6.f.2

   2) Welds covered by paragraphs 4.2.5.6.e.5 and 4.2.5.6.f.6 may be considered integral

4.2.5.7 Special Limitations for Joints In Quenched and Tempered High Strength Steels

a) In vessels and vessel parts constructed of quenched and tempered high strength steels (see Table 3.A.2) except as permitted in paragraph 4.2.5.7.b, all joints of Categories A, B, and C, and all other welded joints between parts of the pressure containing enclosure that are not defined by the category designation shall be Type No.1.

   1) If the shell plate thickness is 50 mm (2 in.) or less, then all Category D welds shall be Type No. 1 in accordance with Table 4.2.13.

   2) If the shell plate thickness is greater than 50 mm (2 in.), then the weld detail may be as permitted for nozzles in Table 4.2.10 or Table 4.2.13.

b) For materials SA-333 Grade 8, SA-334. Grade 8, SA-353, SA-522, SA-553, and SA-645 the weld joints shall be as follows:

   1) All joints of Category A shall be Type No.1.

   2) All joints of Category B shall be Type No.1 or Type No.2.

   3) All joints of Category C shall be full penetration welds extending through the entire section at the joint.

   4) All joints of Category D attaching a nozzle neck to the vessel wall and to a reinforcing pad, if used, shall be full penetration groove welds.

4.2.5.8 Tube-To-Tubesheet Welds

Requirements for tube-to-tubesheet welds are given in paragraph 4.18.

4.2.6 Nomenclature

a	geometry parameter used to determine the length requirements for a thickness transition or a required weld size, applicable.
b	geometry parameter used to determine the length requirements for a thickness transition or a required weld size, applicable.
c	weld size parameter
R	mean radius of the shell
S_{yT}	minimum specified yield strength from Annex 3.D at the design temperature.
S_u	minimum specified ultimate tensile strength from Annex 3.D.
t_a	thickness of the attached member
t_c	throat dimension of a corner weld
t_e	thickness of the reinforcing element
t_h	nominal thickness of the head
t_n	nominal thickness of the shell or nozzle, as applicable
t_p	distance from the outside surface of a flat head, flange, or other part to either the edge or center of a weld.
t_{pipe}	minimum wall thickness of the connecting pipe
t_r	required thickness of the shell in accordance with the requirements of this Division
t_s	nominal thickness of the shell
t_w	depth of penetration of the weld
t_x	two times the thickness g_0 (see paragraph 4.16) when the design is calculated as an integral flange or two times the nozzle thickness of the shell nozzle wall required for internal pressure when the design is calculated as a loose flange, but in no case less than 6 mm (0.25 in.).
T	minimum thickness of a flat head, cover, flange, or tubesheet, as applicable
W_r	weld type reduction factor

4.2.7 Tables

Table 4.2.1 – Definition Of Weld Categories

Weld Category	Description
A	• Longitudinal and spiral welded joints within the main shell, communicating chambers (1), transitions in diameter, or nozzles • Any welded joint within a sphere, within a formed or flat head, or within the side plates (2) of a flat-sided vessel • Circumferential welded joints connecting hemispherical heads to main shells, to transitions in diameter, to nozzles, or to communicating chambers.
B	• Circumferential welded joints within the main shell, communicating chambers (1), nozzles or transitions in diameter including joints between the transition and a cylinder at either the large or small end • Circumferential welded joints connecting formed heads other than hemispherical to main shells, to transitions in diameter, to nozzles, or to communicating chambers.
C	• Welded joints connecting flanges, Van Stone laps, tubesheets or flat heads to main shell, to formed heads, to transitions in diameter, to nozzles, or to communicating chambers (1) • Any welded joint connecting one side plate (2) to another side plate of a flat-sided vessel.
D	• Welded joints connecting communicating chambers (1) or nozzles to main shells, to spheres, to transitions in diameter, to heads, or to flat-sided vessels • Welded joints connecting nozzles to communicating chambers (1) (for nozzles at the small end of a transition in diameter see Category B).
E	• Welded joints attaching nonpressure parts and stiffeners

Notes:

1. Communicating chambers are defined as appurtenances to the vessel that intersect the shell or heads of a vessel and form an integral part of the pressure containing enclosure, e.g., sumps.
2. Side plates of a flat-sided vessel are defined as any of the flat plates forming an integral part of the pressure containing enclosure.

Table 4.2.2 – Definition Of Weld Joint Types

Weld Joint Type	Description
1	Butt joints and angle joints where the cone half-apex angle is less than or equal to 30 degrees produced by double welding or by other means which produce the same quality of deposited weld metal on both inside and outside weld surfaces. Welds using backing strips which remain in place do not qualify as Type No.1 butt joints.
2	Butt joints produced by welding from one side with a backing strip that remains in place.
3	Butt joints produced by welding from one side without a backing strip.
7	Corner joints made with full penetration welds with or without cover fillet welds
8	Angle joints made with a full penetration weld where the cone half-apex angle is greater than 30 degrees
9	Corner joints made with partial penetration welds with or without cover fillet welds
10	Fillet welds

Table 4.2.3 – Definition Of Material Types For Welding And Fabrication Requirements

Material Type	Description
1	• P-No. 1 Groups 1, 2, and 3 • P-No. 3 Group 3 except SA-302 • P-No. 4, Group 1, SA-387 Grade 12 only • P-No. 8, Groups 1 and 2 • P-No. 9A Group 1
2	Materials not included in Material Types 1, 3 and 4
3	Quenched and Tempered High Strength Steels (see Table 3.A.4) except SA-372 Types IV & V when used for Forged Bottles
4	• P-No. 21 through P-No. 25 inclusive • P-No. 31 through P-No. 35 inclusive • P-No. 41 through P-No. 45 inclusive

Table 4.2.4 – Some Acceptable Weld Joints For Shell Seams

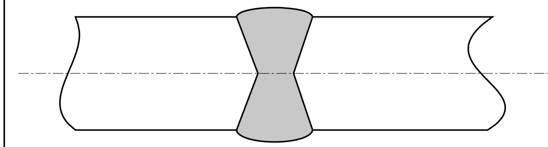
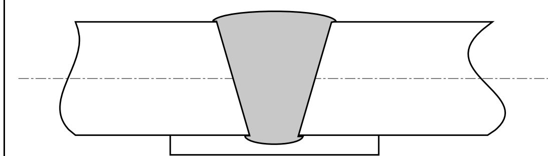
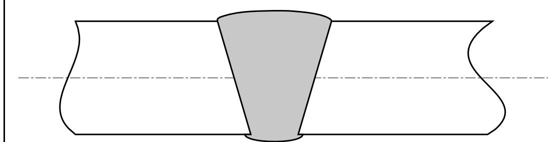
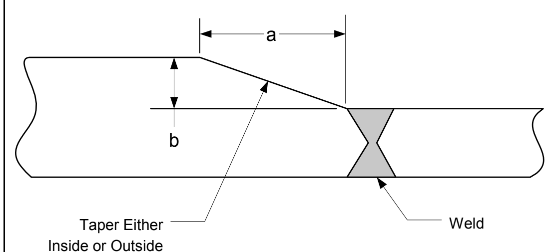
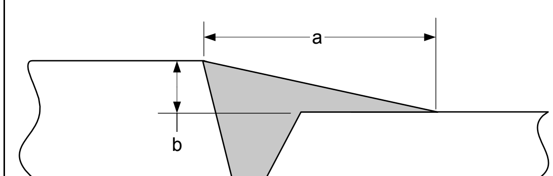
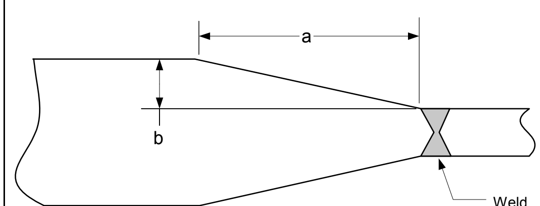
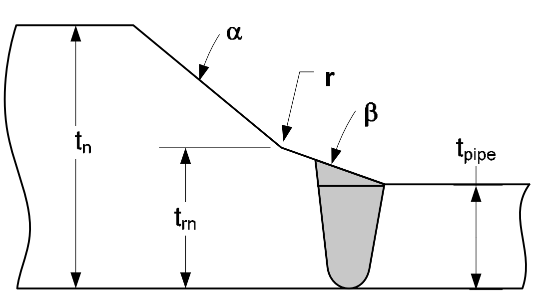
Detail	Joint Type	Joint Category	Design Notes	Figure
1	1	A,B,C,D		
2	2	B		
3	3	B		
4	1	A,B,C,D	• $a \geq 3b$ • The length of the taper, a, may include the weld • Joint Types 2 and 3 may be permissible, see paragraphs 4.2.5.2 through 4.2.5.6 for limitations	
5	1	A,B,C,D		
6	1	A,B,C,D		
7	1	B	• The weld bevel is shown for illustration only • $t_1 \geq \max\left[0.8t_{rn}, t_{pipe}\right]$ • $\alpha \leq 30°$ • $\beta, 14° \leq \beta \leq 18.5°$ • ***r***, 6 mm (0.25 in.) min. radius • Joint Types 2 and 3 may be permissible, see paragraphs 4.2.5.2 through 4.2.5.6 for limitations	

Table 4.2.4 – Some Acceptable Weld Joints For Shell Seams

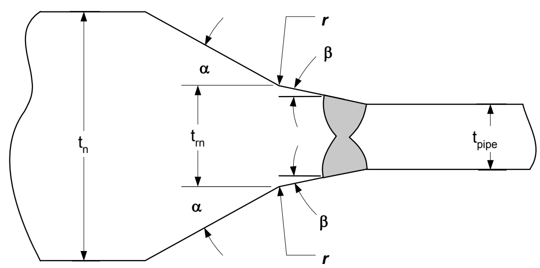
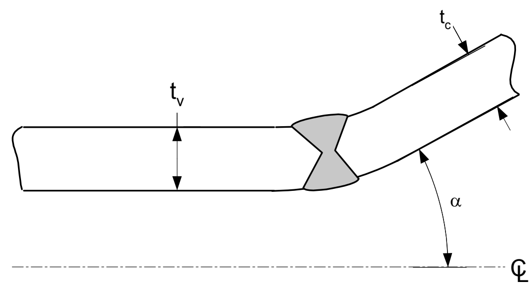
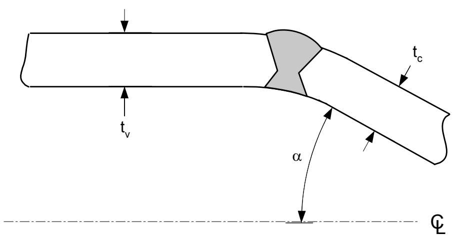
Detail	Joint Type	Joint Category	Design Notes	Figure
8	1	B	• $t_1 \geq \max\left[0.8t_{rn}, t_{pipe}\right]$ • $\alpha \leq 30°$ • $\beta, 14° \leq \beta \leq 18.5°$ • **r**, 6 mm (0.25 in.) min. radius • Joint Types 2 and 3 may be permissible, see paragraphs 4.2.5.2 through 4.2.5.6 for limitations	
9	1	B	• $\alpha \leq 30^0$ • see paragraph 4.2.5.3.f • Joint Types 2 and 3 may be permissible, see paragraphs 4.2.5.2 through 4.2.5.6 for limitations	
10	8	B	• $\alpha > 30^0$	
11	1	B	• $\alpha \leq 30^0$ • see paragraph 4.2.5.3.f • Joint Types 2 and 3 may be permissible, see paragraphs 4.2.5.2 through 4.2.5.6 for limitations	
12	8	B	• $\alpha > 30^0$	

Table 4.2.5 – Some Acceptable Weld Joints For Formed Heads

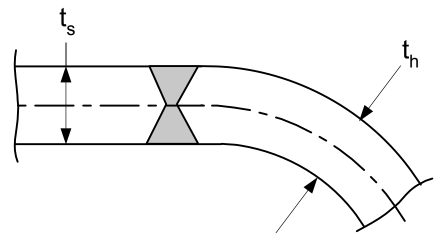
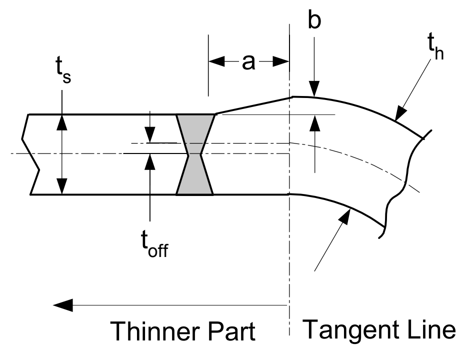
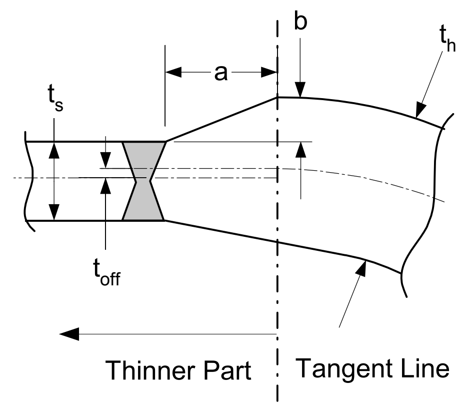
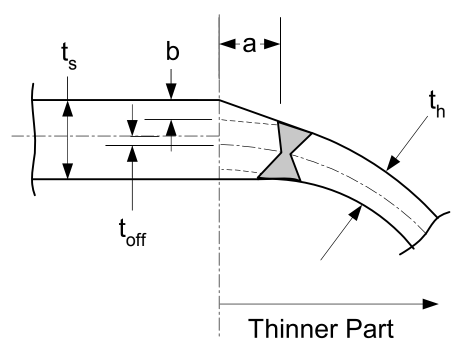
Detail	Joint Type	Joint Category	Design Notes	Figure
1	1	A,B	• Joint Types 2 and 3 may be permissible, see paragraphs 4.2.5.2 through 4.2.5.6 for limitations	
2	1	A,B	• $a \ge 3b$ when t_h exceeds t_s. • $t_{off} \le 0.5(t_h - t_s)$ • The skirt minimum length is $\min[3t_h, 38\,mm\,(1.5in)]$ except when necessary to provide the required taper length • If $t_h \le 1.25t_s$, then the length of the skirt shall be sufficient for any required taper • The length of the taper a may include the width of the weld. • The shell plate center line may be on either side of the head plate center line • Joint Types 2 and 3 may be permissible, see paragraphs 4.2.5.2 through 4.2.5.6 for limitations	
3	1	A,B		
4	1	A,B	• $a \ge 3b$ • $t_{off} \le 0.5(t_h - t_s)$ • The length of the taper a may include the width of the weld. • The shell plate center line may be on either side of the head plate center line • Joint Types 2 and 3 may be permissible, see paragraphs 4.2.5.2 through 4.2.5.6 for limitations	

Table 4.2.5 – Some Acceptable Weld Joints For Formed Heads

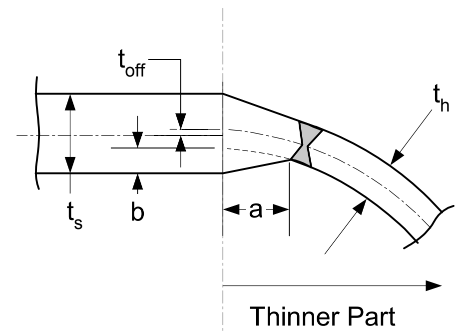
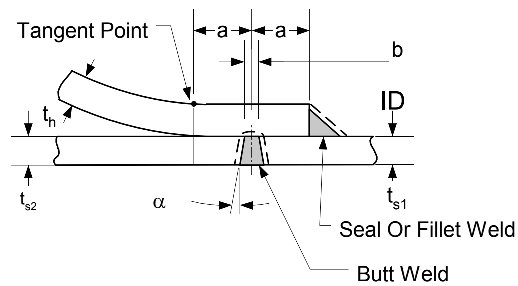
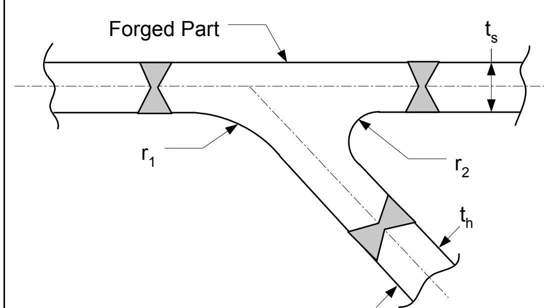
Detail	Joint Type	Joint Category	Design Notes	Figure
5	1	A,B		
6	2	B	• Butt weld and , if used, fillet weld shall be designed to take a shear load at 1.5 times the design differential pressure • $a \geq \min\left[2t_h,\ 25\,mm\,(1\,in)\right]$ • b, 13 mm (0.5 in.) minimum • The shell thicknesses t_{s1} and t_{s2} may be different • $\alpha, 15° \leq \alpha \leq 20°$	
7	1	A,B	• $r_1 \geq 2r_2$ • $r_2 \geq \min\left[t_s,\ t_h\right]$	

Table 4.2.6 – Some Acceptable Weld Joints For Unstayed Flat Heads, Tubesheets Without A Bolting Flange, And Side Plates of Rectangular Pressure Vessels

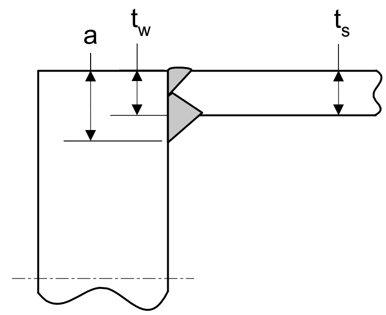
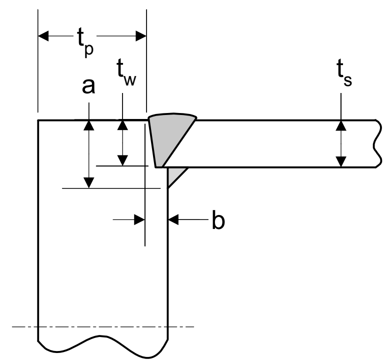
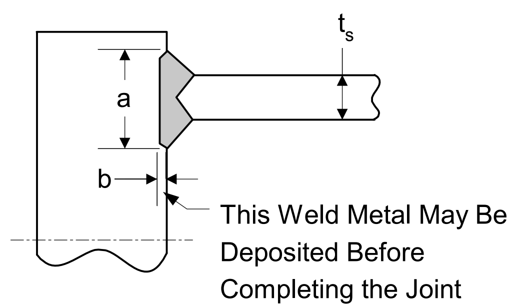
Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	C	• $a \ge 2t_s$ • $t_w \ge t_s$	
2	7	C	• $a + b \ge 2t_s$ • $t_w \ge t_s$ • $t_p \ge \min\left[t_s,\ 6\,mm\,(0.25\,in)\right]$ • The dimension b is produced by the weld preparation and shall be verified after fit-up and before welding	
3	7	C	• $a + b \ge 2t_s$ • $b = 0$ is permissible • The dimension b is produced by the weld preparation and shall be verified after fit-up and before welding	

Table 4.2.7 – Some Acceptable Weld Joints With Butt Weld Hubs

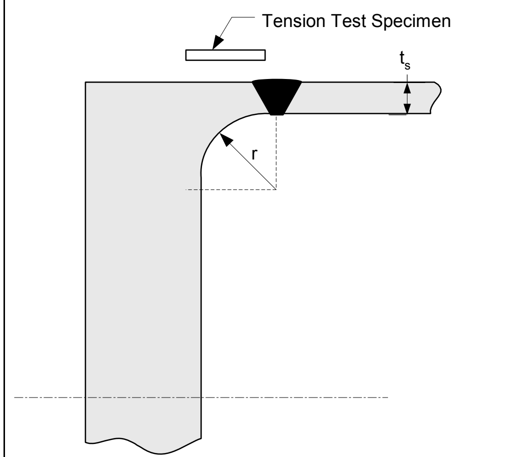
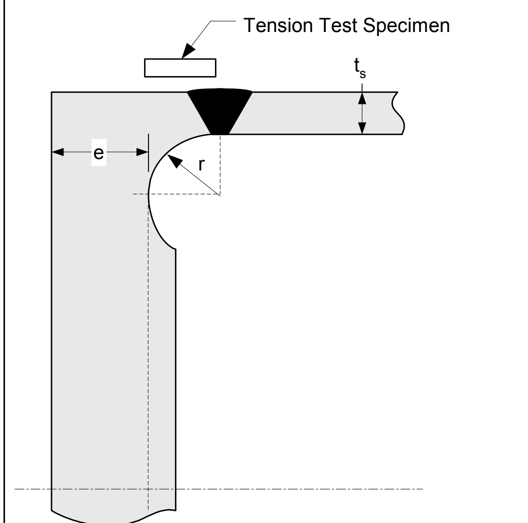
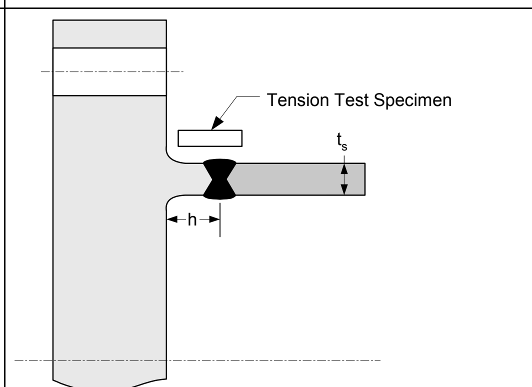
Detail	Joint Type	Joint Category	Design Notes	Figure
1	1	C	• $r \geq 10\,mm\ (0.375\,in.)$ for $t_s \leq 38\,mm\,(1.5\,in)$ • $r \geq \min\left[0.25t_s,\ 19\,mm\,(0.75\,in)\right]$ for $t_s > 38\,mm\,(1.5\,in)$	
2	1	C	• $r \geq 10\,mm\ (0.375\,in.)$ for $t_s \leq 38\,mm\,(1.5\,in)$ • $r \geq \min\left[0.25t_s,\ 19\,mm\,(0.75\,in)\right]$ for $t_s > 38\,mm\,(1.5\,in)$ • $e \geq \max\left[t_s,\ T\right]$	
3	1	C	• $h = \max\left[1.5t_s,\ 19\,mm\,(0.75\,in)\right]$ but need not exceed $51\,mm\,(2\,in)$	

Table 4.2.8 – Some Acceptable Weld Joints For Attachment Of Tubesheets With A Bolting Flange

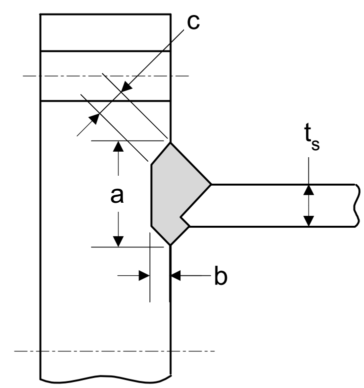
Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	C	• $a + b \ge 2t_s$ • $b = 0$ is permissible • The dimension b is produced by the weld preparation and shall be verified after fit-up and before welding • $c \ge \min\left[0.7t_s, 1.4t_r\right]$	

Table 4.2.9 – Some Acceptable Weld Joints For Flange Attachments

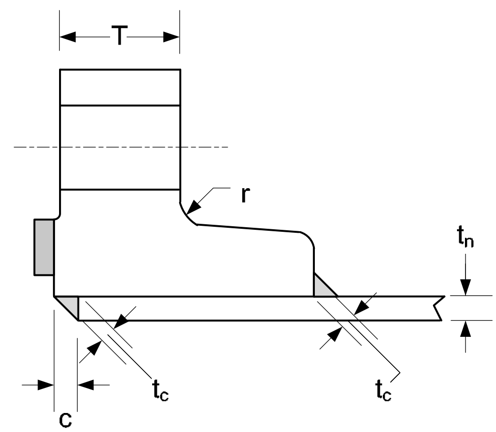
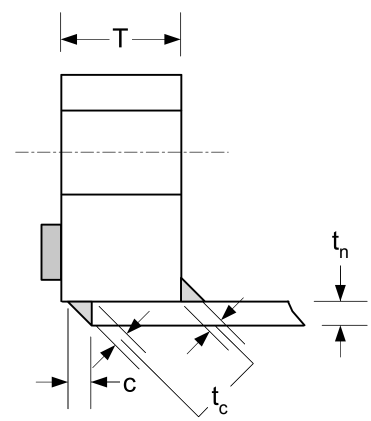
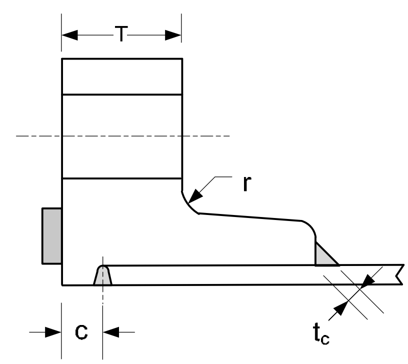
Detail	Joint Type	Joint Category	Design Notes	Figure
1	10	C	• Loose Type Flange • $t_c \geq 0.7t_n$ • $c \leq t_n + 6\,mm\,(0.25\,in)$ maximum • $r \geq \max\left[0.25g_1, 5mm\,(0.1875\,in.)\right]$	
2	10	C	• Loose Type Flange • $t_c \geq 0.7t_n$ • $c \leq t_n + 6\,mm\,(0.25\,in)$ maximum	
3	7	C	• Loose Type Flange • $t_c \geq 0.7t_n$ • $c \leq 0.5t$ maximum • $r \geq \max\left[0.25g_1, 5mm\,(0.1875\,in.)\right]$	

Table 4.2.9 – Some Acceptable Weld Joints For Flange Attachments

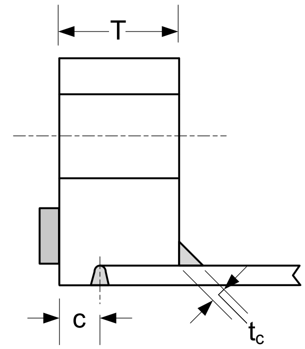
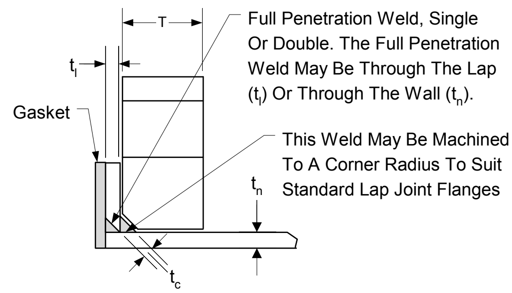
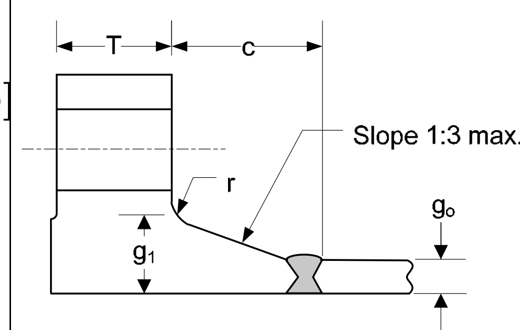
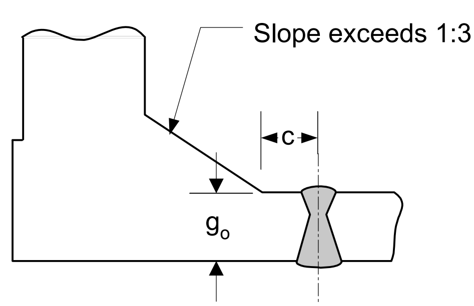
Detail	Joint Type	Joint Category	Design Notes	Figure
4	7	C	• Loose Type Flange • $t_c \geq 0.7t_n$ • $c \leq 0.5t$ maximum	
5	7	C	• Loose Type Flange • $t_c \geq 0.7t_n$ • $t_l \geq t_n + 5 mm (0.1875 in.)$	
6	1	C	• Integral Type Flange • $c \geq 1.5g_0$ minimum • $r \geq \max\left[0.25g_1, 5mm(0.1875 in.)\right]$	
7	1	C	• Integral Type Flange • $c \geq 1.5g_0$ minimum	

Table 4.2.9 – Some Acceptable Weld Joints For Flange Attachments

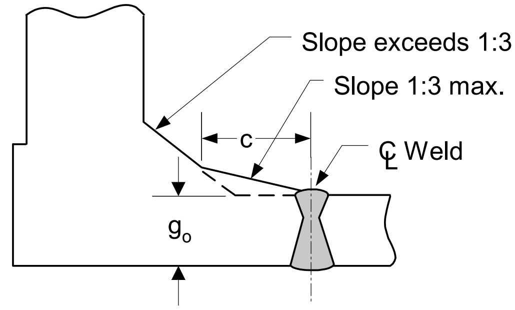
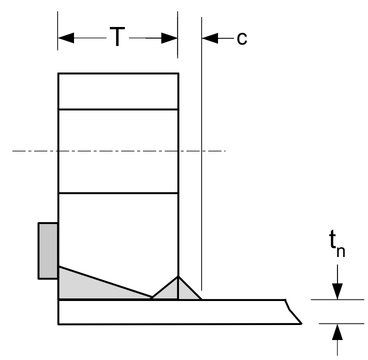
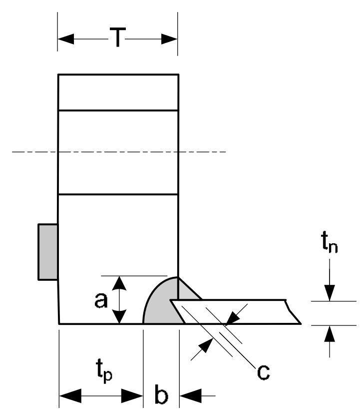
Detail	Joint Type	Joint Category	Design Notes	Figure
8	1	C	• Integral Type Flange • $c \geq 1.5 g_0$ minimum	
9	7	C	• Integral Type Flange • $c \geq \min\left[0.25 g_0,\ 6\,mm\,(0.25\,in)\right]$	
10	7	C	• Integral Type Flange • $a+b \geq 3t_n$ • $t_p \geq \min\left[t_n, 6\ mm\ (0.25\ in.)\right]$ • $c \geq \min\left[t_n, 6\ mm\ (0.25\ in.)\right]$	

Table 4.2.10 – Some Acceptable Full Penetration Welded Nozzle Attachments Not Readily Radiographable

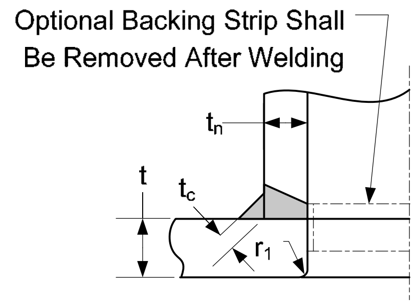
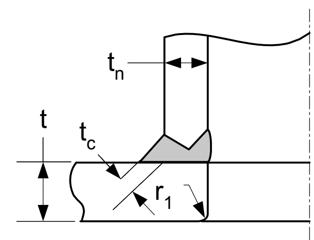
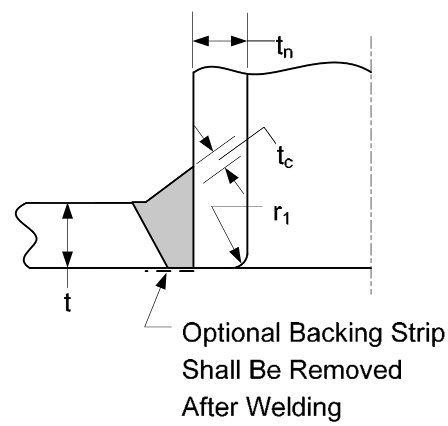
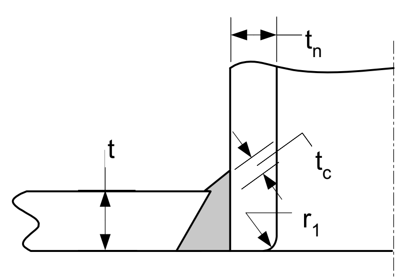
Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	D	• $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $r_1 \geq \min\left[0.25t, 3mm\left(0.125in\right)\right]$	
2	7	D	• $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $r_1 \geq \min\left[0.25t, 3mm\left(0.125in\right)\right]$	
3	7	D	• $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $r_1 \geq \min\left[0.25t, 3mm\left(0.125in\right)\right]$	
4	7	D	• $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $r_1 \geq \min\left[0.25t, 3mm\left(0.125in\right)\right]$	

Table 4.2.10 – Some Acceptable Full Penetration Welded Nozzle Attachments Not Readily Radiographable

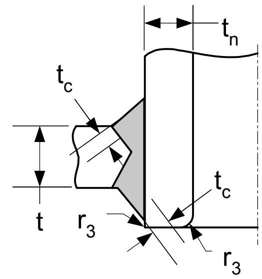
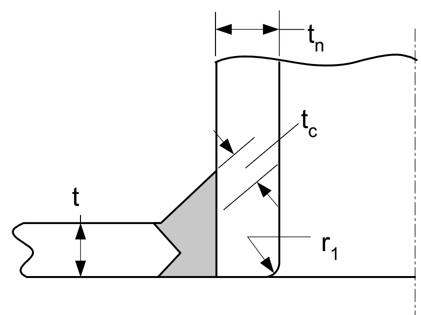
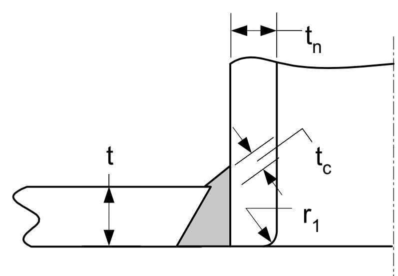
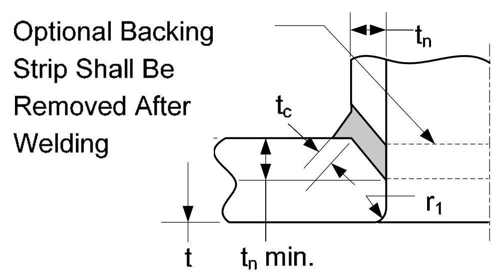
Detail	Joint Type	Joint Category	Design Notes	Figure
5	7	D	• $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $r_3 \geq \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$ alternatively, a chamfer of $r_3 \geq \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$ at 45 degrees	
6	7	D	• $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $r_1 \geq \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$	
7	7	D	• $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $r_1 \geq \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$	
8	7	D	• $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $r_1 \geq \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$	

Table 4.2.10 – Some Acceptable Full Penetration Welded Nozzle Attachments Not Readily Radiographable

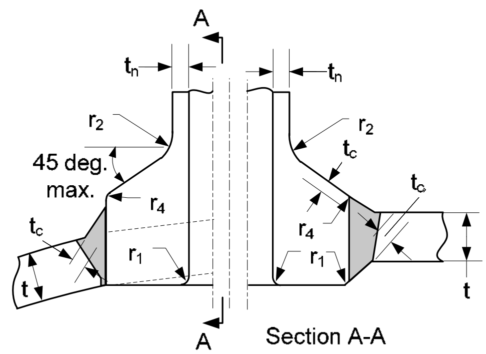
Detail	Joint Type	Joint Category	Design Notes	Figure
9	7	D	• $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $r_1 \geq \min\left[0.25t, 3mm\left(0.125in\right)\right]$ • $r_2 \geq 19\,mm\ (0.75\ in.)$ • $r_4 \geq 6\,mm\ (0.25\ in.)$	Sections Perpendicular and Parallel to the Cylindrical Vessel Axis

Table 4.2.11 – Some Acceptable Pad Welded Nozzle Attachments And Other Connections To Shells

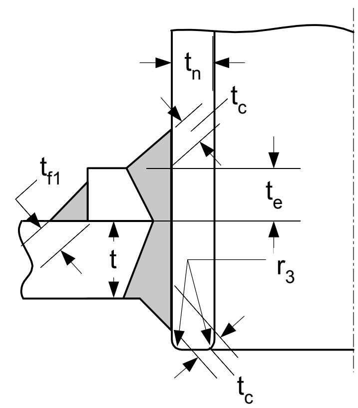
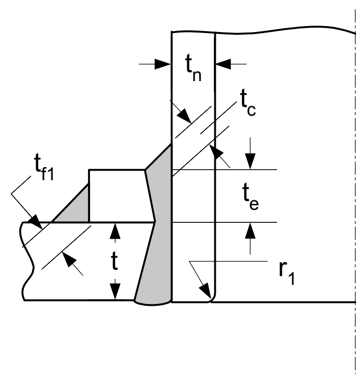
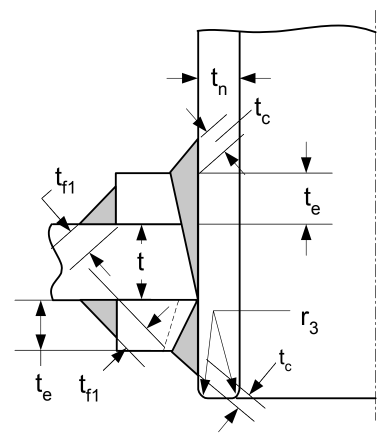
Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	D	• $t_c \ge \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $t_{f1} \ge \min\left[0.6t_e,\ 0.6t\right]$ • $r_3 \ge \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$ alternatively, a chamfer of $r_3 \ge \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$ at 45 degrees	
2	7	D	• $t_c \ge \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $t_{f1} \ge \min\left[0.6t_e,\ 0.6t\right]$ • $r_1 \ge \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$	
3	7	D	• $t_c \ge \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $t_{f1} \ge \min\left[0.6t_e,\ 0.6t\right]$ • $r_3 \ge \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$ alternatively, a chamfer of $r_3 \ge \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$ at 45 degrees	

Table 4.2.11 – Some Acceptable Pad Welded Nozzle Attachments And Other Connections To Shells

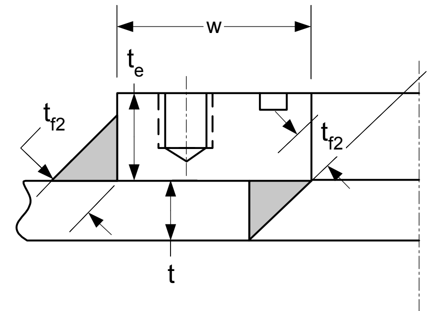
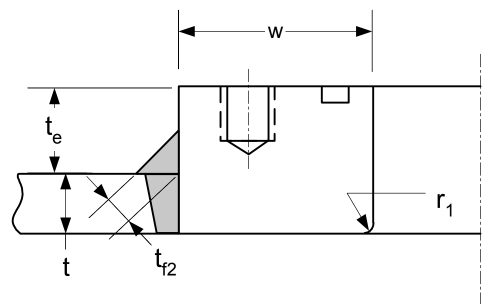
Detail	Joint Type	Joint Category	Design Notes	Figure
4	10	D	• $t_{f2} \geq \min\left[0.7t_e,\ 0.7t\right]$	
5	7	D	• $t_{f2} \geq \min\left[0.7t_e,\ 0.7t\right]$ • $r_1 \geq \min\left[0.25t, 3mm\left(0.125in\right)\right]$	

Table 4.2.12 – Some Acceptable Fitting Type Welded Nozzle Attachments And Other Connections To Shells

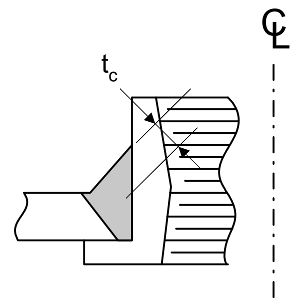
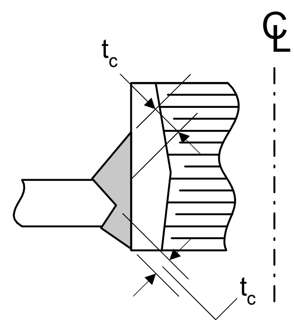
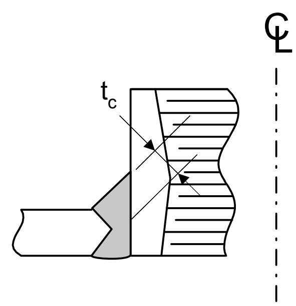
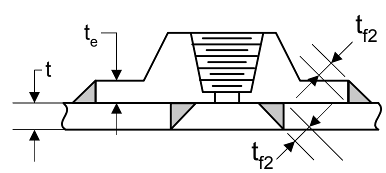
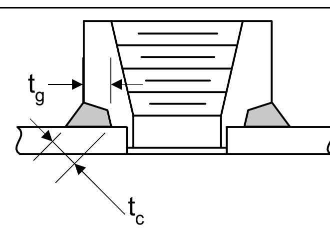
Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	D	• Limited to DIN 50 (NPS 2) and smaller • $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$	
2	7	D	• Limited to DIN 50 (NPS 2) and smaller • $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$	
3	7	D	• Limited to DIN 50 (NPS 2) and smaller • $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$	
4	10	D	• Limited to DIN 50 (NPS 2) and smaller • $t_c \geq \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $t_{f2} \geq \min\left[0.7t_e,\ 0.7t\right]$	
5	9	D	• Limited to DIN 50 (NPS 2) maximum • The groove weld t_g shall not be less than the thickness of Schedule 160 • $t_c \geq \min\left[0.7t_n,\ 6\ mm\left(0.25\ in\right)\right]$	

Table 4.2.13 – Some Acceptable Welded Nozzle Attachments That Are Readily Radiographable

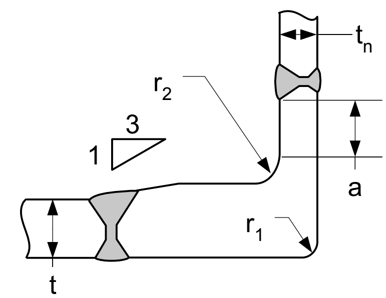
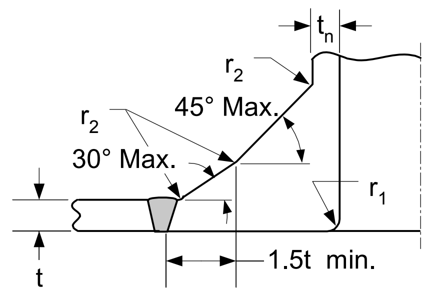
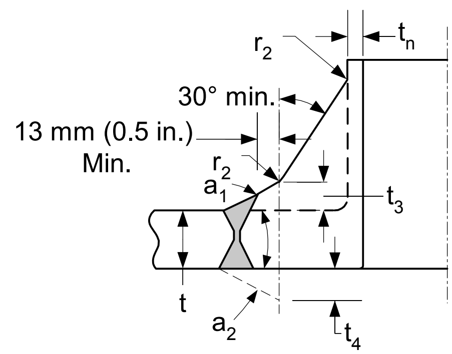
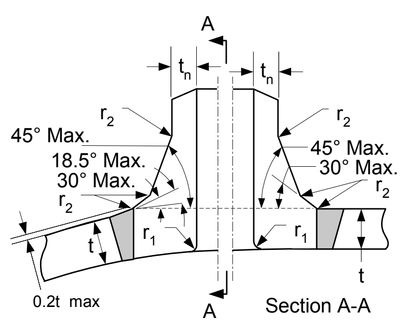
Detail	Joint Type	Joint Category	Design Notes	Figure
1	1	D	• $r_1 \geq \min\left[0.25t, 3\,mm\left(0.125\,in\right)\right]$ • $r_2 \geq \min\left[0.25t_n, 19\,mm\,(0.75\ in)\right]$	
2	1	D	• $r_1 \geq \min\left[0.25t, 3\,mm\left(0.125\,in\right)\right]$ • $r_2 \geq \min\left[0.25t_n, 19\,mm\,(0.75\ in)\right]$	
3	1	D	• $r_1 \geq \min\left[0.25t, 3\,mm\left(0.125\,in\right)\right]$ • $r_2 \geq \min\left[0.25t_n, 19\,mm\,(0.75\ in)\right]$ • $t_3 + t_4 \leq 0.2t$ • $a_1 + a_2 \leq 18.5\ Degree$	
4	1	D	• $r_1 \geq \min\left[0.25t, 3\,mm\left(0.125\,in\right)\right]$ • $r_2 \geq \min\left[0.25t_n, 19\,mm\,(0.75\ in)\right]$	 Sections Perpendicular and Parallel to the Cylindrical Vessel Axis

Table 4.2.13 – Some Acceptable Welded Nozzle Attachments That Are Readily Radiographable

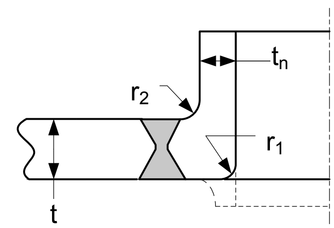
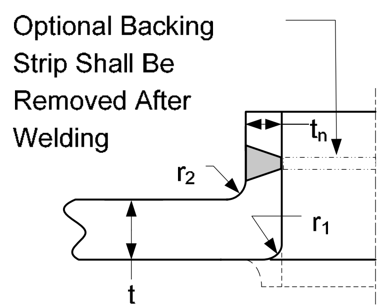
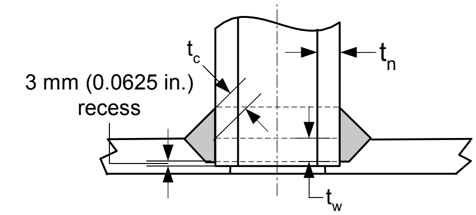
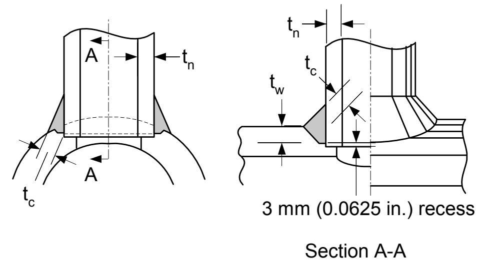
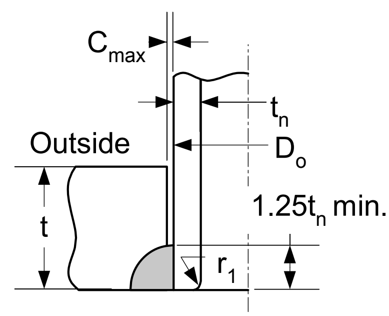
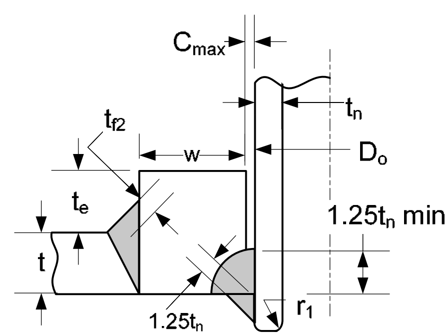
Detail	Joint Type	Joint Category	Design Notes	Figure
5	1	D	• $r_1 \geq \min\left[0.25t_n, 3mm\left(0.125\,in\right)\right]$ $r_2 \geq \min\left[0.25t_n, 19\,mm\,(0.75\ in)\right]$	
6	1	D	• $r_1 \geq \min\left[0.25t_n, 3mm\left(0.125\,in\right)\right]$ $r_2 \geq \min\left[0.25t_n, 19\,mm\,(0.75\ in)\right]$	

Table 4.2.14 – Some Acceptable Partial Penetration Nozzle Attachments

Detail	Joint Type	Joint Category	Design Notes	Figure
1	9	D	• $t_c \ge \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $t_w \ge 1.25t_n$	
2	9	D	• $t_c \ge \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $t_w \ge 1.25t_n$	
3	9	D	• $r_1 \ge \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$ $C_{\max}$ defined as follows: $0.25\,mm,\ D_o \le 25\,mm$ $0.51\,mm,\ 25mm < D_o \le 102\,mm$ • $0.76\,mm,\ D_o > 102\,mm$ $0.01\,in,\ D_o \le 1\,in$ $0.02\,in,\ 1\,in < D_o \le 4\,in$ $0.03\,in,\ D_o > 4\ in$	
4	9	D	• $t_{f2} \ge \min\left[0.7t_e,\ 0.7t\right]$ • $r_1 \ge \min\left[0.25t, 3mm\left(0.125\,in\right)\right]$ $C_{\max}$ defined as follows: $0.25\,mm,\ D_o \le 25\,mm$ $0.51\,mm,\ 25mm < D_o \le 102\,mm$ • $0.76\,mm,\ D_o > 102\,mm$ $0.01\,in,\ D_o \le 1\,in$ $0.02\,in,\ 1\,in < D_o \le 4\,in$ $0.03\,in,\ D_o > 4\ in$	

4.2.8 Figures

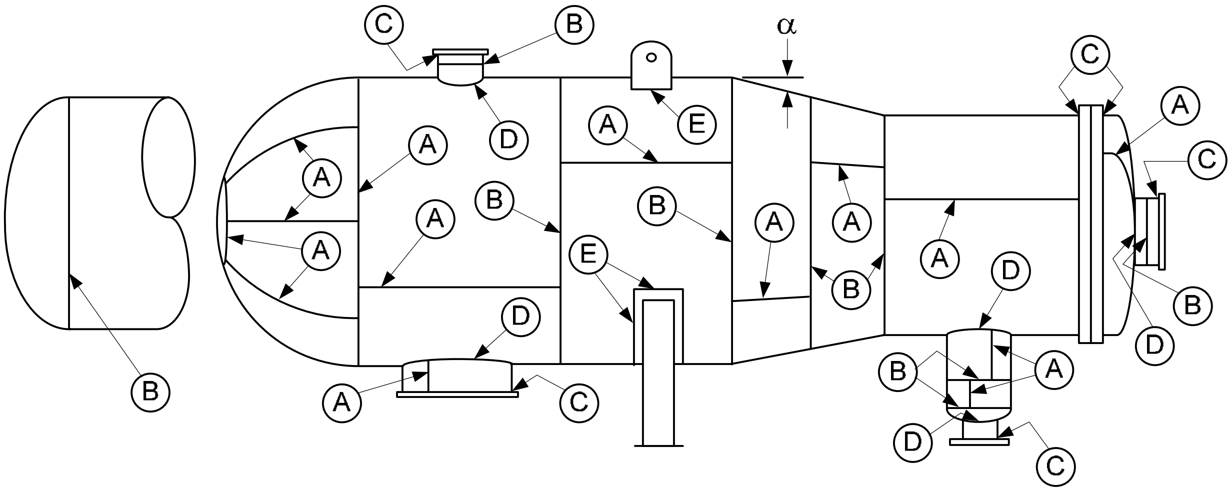
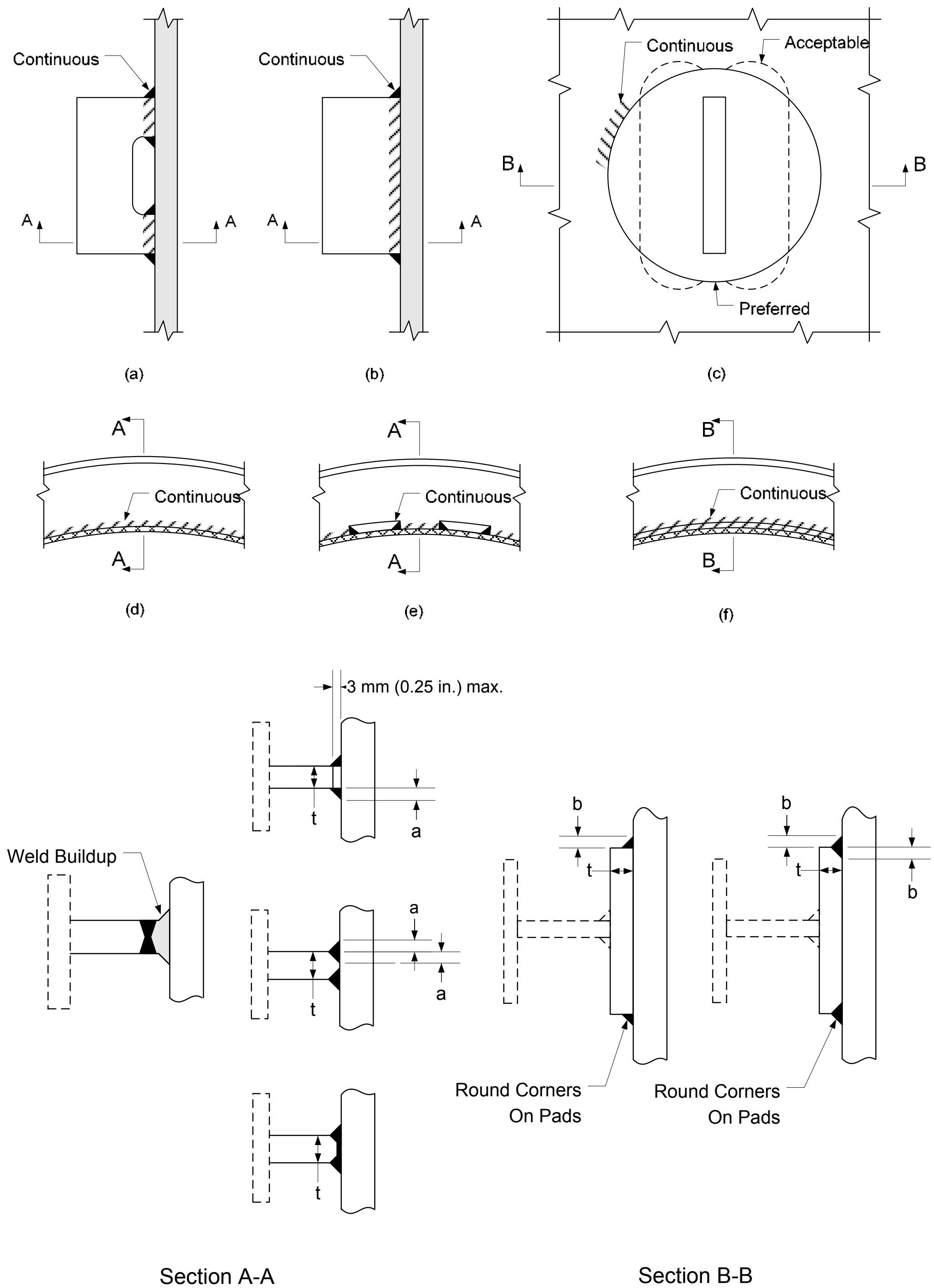


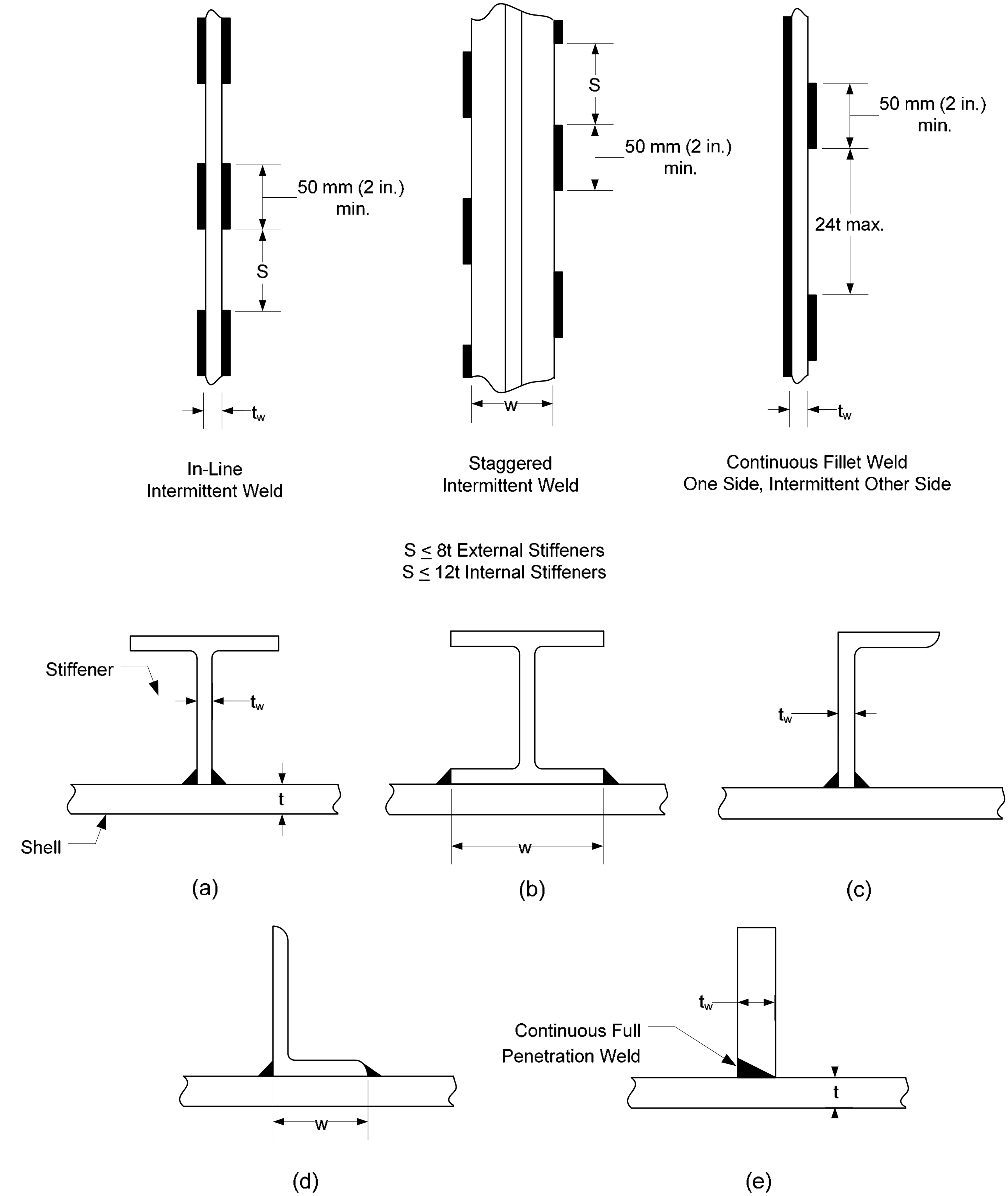
Figure 4.2.1 – Weld Joint Locations Typical Of categories A, B, C, D, and E



Notes:

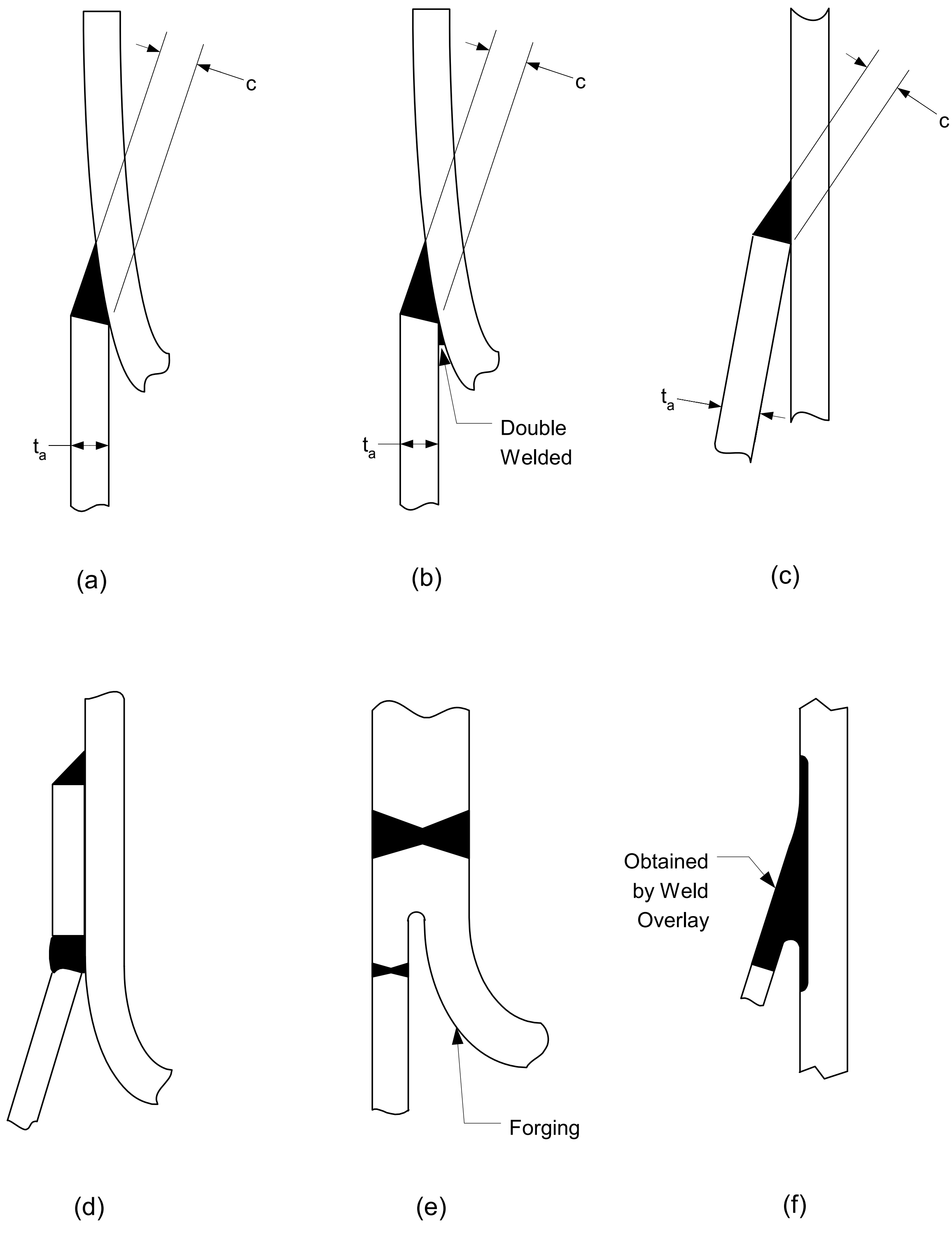
1. Attachment weld size: $a \geq 0.25t$ and $b \geq 0.5t$
2. Vents holes shall be considered for continuously attached pads
3. For design (e) above, a minimum of 50% of the web must be welded, evenly spaced around the circumference of the shell

Figure 4.2.2 – Some Bracket, Lug and Stiffener Attachment Weld Details



Note: see paragraph 4.2.5.6.e for limitations

Figure 4.2.3 – Some Acceptable Methods of Attaching Stiffening Rings



Notes:
1. All welds are continuous
2. c is the minimum thickness of the weld metal from the root to the face of the weld
3. Attachment weld size: $c \geq t_a$

Figure 4.2.4 – Some Acceptable Skirt Weld Details

4.3 Design Rules For Shells Under Internal Pressure

4.3.1 Scope

4.3.1.1 Paragraph 4.3 provides rules for determining the required wall thickness of cylindrical, conical, spherical, torispherical, and ellipsoidal shells and heads subject to internal pressure. In this context, internal pressure is defined as pressure acting on the concave side of the shell.

4.3.1.2 The effects of supplemental loads are not included in design equations for shells and heads included in paragraphs 4.3.3 to 4.3.7. Supplemental loads shall be defined in the User's Design Specification and their effects that result in combined loadings shall be evaluated in a separate analysis performed in accordance with the methods in paragraph 4.3.10.

4.3.1.3 Rules are provided for the design of cylindrical-to-conical shell transition junctions in paragraphs 4.3.11 and 4.3.12. To facilitate the use of these rules, the shell wall thickness and stiffener configuration, as applicable, shall be designed using the rules in paragraphs 4.3.3 through 4.3.7. After an initial design is determined, this design should then be checked and modified as required using the rules of paragraphs 4.3.12 and 4.3.13.

4.3.2 Shell Tolerances

4.3.2.1 The shell of a completed vessel shall satisfy the following requirements.

a) The difference between the maximum and minimum inside diameters at any cross section shall not exceed 1% of the nominal diameter at the cross section under consideration. The diameters may be measured on the inside or outside of the vessel. If measured on the outside, the diameters shall be corrected for the plate thickness at the cross section under consideration.

b) When the cross section passes through an opening or within one inside diameter of the opening measured from the center of the opening, the permissible difference in inside diameters given above may be increased by 2% of the inside diameter of the opening. When the cross section passes through any other location normal to the axis of the vessel, including head-to-shell junctions, the difference in diameters shall not exceed 1%..

4.3.2.2 Tolerances for formed heads shall satisfy the following requirements.

a) The inner surface of torispherical, toriconical, hemispherical, or ellipsoidal heads shall not deviate outside of the specified shape by more than 1.25% of D nor inside the specified shape by more than 0.625% of D, where D is the nominal inside diameter of the vessel shell at the point of attachment. Such deviations shall be measured perpendicular to the specified shape and shall not be abrupt. The knuckle radius shall not be less than that specified

b) Measurements for determining the deviations specified in paragraph 4.3.2.2.a shall be taken from the surface of the base metal and not from welds.

c) When the straight flange of any unstayed formed head is machined to make a lap joint connection to a shell, the thickness shall not be reduced to less than 90% of that required for a blank head or the thickness of the shell at the point of attachment. When so machined, the transition from the machined thickness to the original thickness of the head shall not be abrupt but shall be tapered for a distance of at least three times the difference between the thicknesses.

4.3.2.3 Shells that do not meet the tolerance requirements of this paragraph may be evaluated using paragraph 4.14.

4.3.3 Cylindrical Shells

4.3.3.1 Required Thickness – The minimum required thickness of a cylindrical shell subjected to internal pressure shall be determined using the following equation.

$$t=\frac{D}{2}\left(\exp\left[\frac{P}{SE}\right]-1\right) \qquad (4.3.1)$$

4.3.3.2 Combined Loadings – cylindrical shells subject to internal pressure and other loadings shall satisfy the requirements of paragraph 4.3.10.

4.3.4 Conical Shells

4.3.4.1 Required Thickness – the minimum required thickness of a conical shell (see Figure 4.3.1) subjected to internal pressure shall be determined using the following equation.

$$t=\frac{D}{2\cos\left[\alpha\right]}\left(\exp\left[\frac{P}{SE}\right]-1\right) \qquad (4.3.2)$$

4.3.4.2 Offset Transitions – the cylinders for an offset cone shall have parallel centerlines that are offset from each other by a distance no greater than the difference of their minimum radii, as shown in Figure 4.3.2. Configurations that do not satisfy this requirement shall be evaluated per Part 5. The offset cone shall be designed as a concentric cone using the angle, α, as defined in Equation (4.3.3).

$$\alpha=\max\left[\alpha_1,\ \alpha_2\right] \qquad (4.3.3)$$

4.3.4.3 Combined Loadings – conical shells subject to external pressure and other loadings shall satisfy the requirements of paragraph 4.3.10.

4.3.5 Spherical Shells and Hemispherical Heads

4.3.5.1 The minimum required thickness of spherical shells and hemispherical heads shall be determined using the following equation:

$$t=\frac{D}{2}\left(\exp\left[\frac{0.5P}{SE}\right]-1\right) \qquad (4.3.4)$$

4.3.5.2 Combined Loadings – spherical shells and hemispherical heads subject to internal pressure and other loadings shall satisfy the requirements of paragraph 4.3.10.

4.3.6 Torispherical Heads

4.3.6.1 Torispherical Heads With The Same Crown and Knuckle Thicknesses – the minimum required thickness of a torispherical head (see Figure 4.3.3) subjected to internal pressure shall be calculated using the following procedure.

a) STEP 1 – Determine the inside diameter, D, and assume values for the crown radius, L, the knuckle radius, r, and the wall thickness t.

b) STEP 2 – Compute the head L/D, r/D, and L/t ratios and determine if the following equations are satisfied. If the equations are satisfied, then proceed to Step 3; otherwise, the head shall be designed in accordance with Part 5.

$$0.7 \le \frac{L}{D} \le 1.0 \tag{4.3.5}$$

$$\frac{r}{D} \ge 0.06 \tag{4.3.6}$$

$$20 \le \frac{L}{t} \le 2000 \tag{4.3.7}$$

c) STEP 3 – Calculate the following geometric constants:

$$\beta_{th} = \arccos\left[\frac{0.5D - r}{L - r}\right], \; radians \tag{4.3.8}$$

$$\phi_{th} = \frac{\sqrt{Lt}}{r}, \; radians \tag{4.3.9}$$

$$R_{th} = \frac{0.5D - r}{\cos\left[\beta_{th} - \phi_{th}\right]} + r \qquad for \quad \phi_{th} < \beta_{th} \tag{4.3.10}$$

$$R_{th} = 0.5D \qquad for \quad \phi_{th} \ge \beta_{th} \tag{4.3.11}$$

d) STEP 4 – Compute the coefficients C_1 and C_2 using the following equations.

$$C_1 = 9.31\left(\frac{r}{D}\right) - 0.086 \qquad for \; \frac{r}{D} \le 0.08 \tag{4.3.12}$$

$$C_1 = 0.692\left(\frac{r}{D}\right) + 0.605 \qquad for \; \frac{r}{D} > 0.08 \tag{4.3.13}$$

$$C_2 = 1.25 \qquad for \; \frac{r}{D} \le 0.08 \tag{4.3.14}$$

$$C_2 = 1.46 - 2.6\left(\frac{r}{D}\right) \qquad for \; \frac{r}{D} > 0.08 \tag{4.3.15}$$

e) STEP 5 – Calculate the value of internal pressure expected to produce elastic buckling of the knuckle.

$$P_{eth} = \frac{C_1 E_T t^2}{C_2 R_{th}\left(\dfrac{R_{th}}{2} - r\right)} \tag{4.3.16}$$

f) STEP 6 – Calculate the value of internal pressure that will result in a maximum stress in the knuckle equal to the material yield strength.

$$P_y = \frac{C_3 t}{C_2 R_{th}\left(\frac{R_{th}}{2r} - 1\right)} \quad (4.3.17)$$

If the allowable stress at the design temperature is governed by time-independent properties, then C_3 is the material yield strength at the design temperature, or $C_3 = S_y$. If the allowable stress at the design temperature is governed by time-dependent properties, then C_3 is determined as follows.

1) If the allowable stress is established based on 90% yield criterion, then C_3 is the material allowable stress at the design temperature multiplied by 1.1, or $C_3 = 1.1S$.

2) If the allowable stress is established based on 67% yield criterion, then C_3 is the material allowable stress at the design temperature multiplied by 1.5, or $C_3 = 1.5S$.

g) STEP 7 – Calculate the value of internal pressure expected to result in a buckling failure of the knuckle.

$$P_{ck} = 0.6P_{eth} \qquad for \quad G \le 1.0 \quad (4.3.18)$$

$$P_{ck} = \left(\frac{0.77508G - 0.20354G^2 + 0.019274G^3}{1 + 0.19014G - 0.089534G^2 + 0.0093965G^3}\right)P_y \qquad for \quad G > 1.0 \quad (4.3.19)$$

where

$$G = \frac{P_{eth}}{P_y} \quad (4.3.20)$$

h) STEP 8 – Calculate the allowable pressure based on a buckling failure of the knuckle.

$$P_{ak} = \frac{P_{ck}}{1.5} \quad (4.3.21)$$

i) STEP 9 – Calculate the allowable pressure based on rupture of the crown.

$$P_{ac} = \frac{2SE}{\frac{L}{t} + 0.5} \quad (4.3.22)$$

j) STEP 10 – Calculate the maximum allowable internal pressure.

$$P_a = \min\left[P_{ak}, P_{ac}\right] \quad (4.3.23)$$

k) STEP 11 – If the allowable internal pressure computed from STEP 10 is greater than or equal to the design pressure, then the design is complete. If the allowable internal pressure computed from STEP 10 is less than the design pressure, then increase the head thickness and repeat Steps 2 through 10. This process is continued until an acceptable design is achieved.

4.3.6.2 Torispherical Heads With Different Crown and Knuckle Thicknesses – a torispherical head formed from several welded components as shown in Figure 4.3.4 may have a smaller thickness in the spherical crown than in the knuckle region. The transition in thickness shall be located on the inside surface of the thicker part, and shall have a taper not exceeding 1:3.

a) The minimum required thickness of the spherical dome of the head shall be determined in accordance with paragraph 4.3.5.

b) The minimum required thickness of the knuckle region of the head shall be determined in accordance with paragraph 4.3.6.1.b.

4.3.6.3 Combined Loadings – torispherical heads subject to internal pressure and other loadings shall satisfy the requirements of paragraph 4.3.10. In this calculation, the torispherical head shall be approximated as an equivalent spherical shell with a radius equal to L.

4.3.7 Ellipsoidal Heads

4.3.7.1 Required Thickness – The minimum required thickness of an ellipsoidal head (see Figure 4.3.5) subjected to internal pressure shall be calculated using the equations in paragraph 4.3.6 with the following substitutions for r and L.

$$r = D\left(\frac{0.5}{k} - 0.08\right) \tag{4.3.24}$$

$$L = D\left(0.44k + 0.02\right) \tag{4.3.25}$$

where

$$k = \frac{D}{2h} \tag{4.3.26}$$

The rules in this paragraph are applicable for elliptical heads that satisfy Equation (4.3.27). Elliptical heads that do not satisfy this equation shall be designed using Part 5.

$$1.7 \le k \le 2.2 \tag{4.3.27}$$

4.3.7.2 Combined Loadings – ellipsoidal heads subject to internal pressure and other loadings shall satisfy the requirements of paragraph 4.3.10. In this calculation, the ellipsoidal head shall be approximated as an equivalent spherical shell with a radius equal to L.

4.3.8 Local Thin Areas

4.3.8.1 Local Thin Areas

Rules for the evaluation of Local Thin Areas are covered in paragraph 4.14.

4.3.8.2 Local Thin Band in Cylindrical Shells

A complete local circumferential band of reduced thickness at a weld joint in a cylindrical shell as shown in Figure 4.3.6 is permitted providing all of the following requirements are met.

a) The design of the local reduced thickness band is evaluated by limit load or elastic plastic analysis in accordance with Part 5. All other applicable requirements of Part 5 for stress analysis and fatigue analysis shall be satisfied.

b) The cylinder geometry shall satisfy $R_m/t \ge 10$.

c) The thickness of the reduced shell region shall not be less than two-thirds of the cylinder required thickness determined in accordance with paragraph 4.3.3.

d) The reduced thickness region shall be on the outside of the vessel shell with a minimum taper transition of 3:1 in the base metal. The transition between the base metal and weld shall be designed to minimize stress concentrations.

e) The total longitudinal length of each local thin region shall not exceed $\sqrt{R_m t}$ (see Figure 4.3.6).

f) The minimum longitudinal distance from the thicker edge of the taper to an adjacent structural discontinuity shall be the greater of $2.5\sqrt{R_m t}$ or the distance required to assure that overlapping of areas where the primary membrane stress intensity exceeds $1.1S$ does not occur.

4.3.9 Drilled Holes Not Penetrating Through the Vessel Wall

4.3.9.1 Design requirements for partially drilled holes that do not penetrate completely through the vessel wall are provided in this paragraph. These rules are not applicable for studded connections or telltale holes.

4.3.9.2 Partially drilled radial holes in cylindrical and spherical shells may be used provided the following requirements are satisfied.

a) The drilled hole diameter is less than or equal to 50 mm (2 in.).

b) The shell inside diameter to thickness ratio is greater than or equal to 10.

c) The centerline distance between any two partially drilled holes or between a partially drilled hole and an unreinforced opening shall satisfy the requirements of paragraph 4.5.13.

d) Partially drilled holes shall not be placed within the limits of reinforcement of a reinforced opening.

e) The outside edge of the hole shall be chamfered. For flat bottom holes, the inside bottom corner of the hole shall have a minimum radius, r_{hr} of the following:

$$r_{hr} = \min\left[\frac{d}{4},\ 6\,mm\ (0.25\ in.)\right] \tag{4.3.28}$$

f) The minimum acceptable remaining wall thickness, t_{rw} at the location of a partially drilled hole shall be determined as follows:

$$t_{rw} \geq \max\left[t_{rw1},\ 6\,mm\,(0.25\ in.)\right] \tag{4.3.29}$$

where,

$$t_{rw1} = t\left(-1.2261727 + 1.9842895\left(\frac{d}{D}\right) - 2.236553\left(\frac{d}{D}\right)^{0.5}\ln\left[\frac{d}{D}\right]\right) \tag{4.3.30}$$

g) The calculated average shear stress, as determined below shall not exceed $0.8S$.

$$\tau_{pd} = \frac{Pd}{4t_{rw}} \tag{4.3.31}$$

4.3.10 Combined Loadings and Allowable Stresses

4.3.10.1 General - The rules of this paragraph shall be used to determine the thickness requirements for cylindrical, spherical, and conical shells subjected to internal pressure plus supplemental loads of applied net section axial force, bending moment, and torsional moment, as shown in Figure 4.3.7. These rules are applicable if the requirements shown below are satisfied. If all of these requirements are not satisfied, the shell section shall be designed per Part 5.

a) The rules are applicable for regions of shells that are $2.5\sqrt{Rt}$ from any major structural discontinuity.

b) These rules do not take into account the action of shear forces, since these loads generally can be disregarded.

c) The ratio of the shell inside radius to thickness is greater than 3.0.

4.3.10.2 The following procedure shall be used to design cylindrical, spherical, and conical shells subjected to internal pressure plus supplemental loads of applied net section axial force, bending moment, and torsional moment is shown below.

a) STEP 1 – Calculate the membrane stress.

1) For cylindrical shells:

$$\sigma_{\theta m} = \frac{P}{E \cdot \ln\left[\dfrac{D_o}{D}\right]} \tag{4.3.32}$$

$$\sigma_{sm} = \frac{1}{E}\left(\frac{PD^2}{D_o^2 - D^2} + \frac{4F}{\pi\left(D_o^2 - D^2\right)} + \frac{32MD_o\cos[\theta]}{\pi\left(D_o^4 - D^4\right)}\right) \tag{4.3.33}$$

$$\tau = \frac{16M_tD_o}{\pi\left(D_o^4 - D^4\right)} \tag{4.3.34}$$

2) For spherical shells:

$$\sigma_{\theta m} = \frac{P}{2E \cdot \ln\left[\dfrac{D_o}{D_i}\right]} \tag{4.3.35}$$

$$\sigma_{sm} = \frac{1}{E}\left(\frac{P}{2 \cdot \ln\left[\dfrac{D_o}{D}\right]} + \frac{4F}{\pi\left(D_o^2 - D^2\right)\sin^2[\phi]} + \frac{32MD_o\cos[\theta]}{\pi\left(D_o^4 - D^4\right)\sin^3[\phi]}\right) \tag{4.3.36}$$

$$\tau = \frac{32MD_o}{\pi\left(D_o^4 - D^4\right)}\frac{\cos[\phi]}{\sin^3[\phi]}\sin[\theta] + \frac{16M_tD_o}{\pi\left(D_o^4 - D^4\right)\sin^2[\phi]} \tag{4.3.37}$$

3) For conical shells:

$$\sigma_{\theta m} = \frac{P}{E \cdot \ln\left[\left(\frac{D_o}{D} - 1\right)\cos[\alpha] + 1\right]} \tag{4.3.38}$$

$$\sigma_{sm} = \frac{1}{E}\left(\frac{PD^2}{\left(D_o^2 - D^2\right)\cos[\alpha]} + \frac{4F}{\pi\left(D_o^2 - D^2\right)\cos[\alpha]} + \frac{32MD_o\cos[\theta]}{\pi\left(D_o^4 - D^4\right)\cos[\alpha]}\right) \tag{4.3.39}$$

$$\tau = \frac{32MD_o}{\pi\left(D_o^4 - D^4\right)}\tan[\alpha]\sin[\theta] + \frac{16M_tD_o}{\pi\left(D_o^4 - D^4\right)} \tag{4.3.40}$$

b) STEP 2 – Calculate the principal stresses.

$$\sigma_1 = 0.5\left(\sigma_{\theta m} + \sigma_{sm} + \sqrt{\left(\sigma_{\theta m} - \sigma_{sm}\right)^2 + 4\tau^2}\right) \tag{4.3.41}$$

$$\sigma_2 = 0.5\left(\sigma_{\theta m} + \sigma_{sm} - \sqrt{\left(\sigma_{\theta m} - \sigma_{sm}\right)^2 + 4\tau^2}\right) \tag{4.3.42}$$

$$\sigma_3 = -0.5P \tag{4.3.43}$$

c) STEP 3 – At any point on the shell, the following limit shall be satisfied.

$$\frac{1}{\sqrt{2}}\left[\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2\right]^{0.5} \le S \tag{4.3.44}$$

d) STEP 4 – For cylindrical and conical shells, if the meridional stress σ_{sm} is compressive, then Equation (4.3.45) shall be satisfied where F_{xa} is evaluated using paragraph 4.4.12.2 with $\lambda = 0.15$. For spherical shells, the allowable compressive stress criteria in paragraph 4.4.12.4 shall be satisfied. Note that the controlling condition for this case may be the combined loadings without internal pressure.

$$\sigma_{sm} \le F_{xa} \tag{4.3.45}$$

4.3.11 Cylindrical-To-Conical Shell Transition Junctions Without a Knuckle

4.3.11.1 The following rules are applicable for the design of conical transitions or circular cross-sections that do not have a knuckle at the large end or flare at the small end under loadings of internal pressure and applied net section axial force and bending moment. Acceptable conical transition details are shown in Figure 4.3.8. Design rules for a knuckle at the large end or flare at the small end are provided in paragraph 4.3.12.

4.3.11.2 Design rules are provided for the cylinder-to-cone junction details shown in Figure 4.3.9. Details with a stiffening ring at the cylinder-to-cone junction, or other details that differ from the ones shown in this figure shall be designed in accordance with Part 5.

4.3.11.3 The length of the conical shell, measured parallel to the surface of the cone shall be equal to or greater than the following value.

$$L_C \geq 2.0\sqrt{\frac{R_L t_C}{\cos[\alpha]}} + 1.4\sqrt{\frac{R_S t_C}{\cos[\alpha]}} \tag{4.3.46}$$

4.3.11.4 The procedure that shall be used to design the large end of a cylinder-to-cone junction without a knuckle is described below.

a) STEP 1 – Compute the large end cylinder thickness, t_L, using paragraph 4.3.3.

b) STEP 2 – Determine the cone half-apex angle, α, and compute the cone thickness, t_C, at the large end using paragraph 4.3.4.

c) STEP 3 – Proportion the cone geometry such that Equation (4.3.46) and the following equations are satisfied. If all of these equations are not satisfied, then the cylinder-to-cone junction shall be designed in accordance with Part 5. In the calculations, if $0° < \alpha \leq 10°$, then use $\alpha = 10°$.

$$20 \leq \frac{R_L}{t_L} \leq 500 \tag{4.3.47}$$

$$1 \leq \frac{t_C}{t_L} \leq 2 \tag{4.3.48}$$

$$\alpha \leq 60° \tag{4.3.49}$$

d) STEP 4 – Determine the net section axial force, F_L, and bending moment, M_L, applied to the conical transition. The thrust load due to pressure shall not be included as part of the axial force, F_L. Determine an equivalent, X_L, using Equation (4.3.50).

$$X_L = \frac{F_L}{2\pi R_L} \pm \frac{M_L}{\pi R_L^2} \tag{4.3.50}$$

e) STEP 5 – Compute the junction transition design parameters. For calculated values of n other than those presented in Table 4.3.3 and Table 4.3.4, linear interpolation of the equation coefficients, C_i, is permitted.

$$n = \frac{t_C}{t_L} \tag{4.3.51}$$

$$H = \sqrt{\frac{R_L}{t_L}} \tag{4.3.52}$$

$$B = \tan[\alpha] \tag{4.3.53}$$

f) STEP 6 – Compute the stresses in the cylinder and cone at the junction using the equations in Table 4.3.1. The allowable stress criterion for a tensile stress is provided in Table 4.3.1. If either the hoop membrane stress, $\sigma_{\theta m}$, or axial membrane stress, σ_{sm}, at the junction is compressive, then the condition of local buckling shall be considered. Local buckling is not a concern if the limits given in Equations (4.3.54) and (4.3.55) are satisfied. F_{ha} is evaluated using paragraph 4.4.5.1, but substituting $F_{he} = 0.4E\left(\frac{t}{D_o}\right)$. F_{xa} is evaluated using paragraph 4.4.12.2.b with $\lambda = 0.15$. If the stresses of the acceptance criteria are satisfied, the design of the junction is complete.

$$\sigma_{\theta m} \le F_{ha} \tag{4.3.54}$$

$$\sigma_{sm} \le F_{xa} \tag{4.3.55}$$

g) STEP 7 – If the stress acceptance criterion in STEP 6 is satisfied, then the design is complete. If the stress acceptance criterion in STEP 6 is not satisfied, the cylinder thickness or cone thickness near the junction may be increased until the stress acceptance criterion is satisfied. The section of increased thickness for the cylinder and cone shall extend a minimum distance from the junction as shown in Figure 4.3.9. Proceed to STEP 3 to repeat the calculation with the new wall thickness.

4.3.11.5 The procedure that shall be used to design the small end of a cylinder-to-cone junction without a flare is described below.

a) STEP 1 – Compute the small end cylinder thickness, t_S, using paragraph 4.3.3.

b) STEP 2 – Determine the cone half-apex angle, α, and compute the cone thickness, t_C, at the small end using paragraph 4.3.4.

c) STEP 3 – Proportion the cone geometry such that Equation (4.3.46) and the following equations are satisfied. If all of these equations are not satisfied, then the cylinder-to-cone junction shall be designed in accordance with Part 5. In the calculations, if $0° < \alpha \le 10°$, then use $\alpha = 10°$.

$$20 \le \frac{R_S}{t_S} \le 500 \tag{4.3.56}$$

$$1 \le \frac{t_C}{t_S} \le 2 \tag{4.3.57}$$

$$\alpha \le 60° \tag{4.3.58}$$

d) STEP 4 – Determine the net section axial force, F_S, and bending moment, M_S, applied to the conical transition. The thrust load due to pressure shall not be included as part of the axial force, F_S. Determine an equivalent, X_S, line using Equation (4.3.59).

$$X_S = \frac{F_S}{2\pi R_S} \pm \frac{M_S}{\pi R_S^2} \tag{4.3.59}$$

e) STEP 5 – Compute the junction transition design parameters. For calculated values of n other than those presented in Table 4.3.5 and Table 4.3.6, linear interpolation of the equation coefficients, C_i, is permitted.

$$n = \frac{t_C}{t_S} \tag{4.3.60}$$

$$H = \sqrt{\frac{R_S}{t_S}} \tag{4.3.61}$$

$$B = \tan\left[\alpha\right] \tag{4.3.62}$$

f) STEP 6 – Compute the stresses in the cylinder and cone at the junction using the equations in Table 4.3.2. The allowable stress criterion for a tensile stress is provided in Table 4.3.2. If either the hoop membrane stress, $\sigma_{\theta m}$, or axial membrane stress, σ_{sm}, at the junction is compressive, then the condition of local bucking shall be considered. Local buckling is not of concern if the limits given in Equations (4.3.54) and (4.3.55) are satisfied, using the procedure provided in paragraph 4.3.11.4.f. If the stresses of the acceptance criteria are satisfied, the design of the junction is complete.

g) STEP 7 – If the stress acceptance criterion in STEP 6 is satisfied, then the design is complete. If the stress acceptance criterion in STEP 6 is not satisfied, the cylinder thickness or cone thickness near the junction may be increased until the stress acceptance criterion is satisfied. The section of increased thickness for the cylinder and cone shall extend a minimum distance from the junction as shown in Figure 4.3.9. Proceed to STEP 3 to repeat the calculation with the new wall thickness.

4.3.12 Cylindrical-To-Conical Shell Transition Junctions with a Knuckle

4.3.12.1 General – The following rules are applicable for the design of conical transitions of circular cross-section with a knuckle at the large end or flare at the small end under loadings of internal pressure and applied net section axial force and bending moment. Acceptable conical transition details are shown in Figure 4.3.10. Design rules for transition junctions without a knuckle at the large end or flare at the small end are provided in paragraph 4.3.11.

4.3.12.2 The procedure that shall be used to design the large end of a cylinder-to-cone junction with a knuckle is described below.

a) STEP 1 – Compute the large end cylinder thickness, t_L, using paragraph 4.3.3.

b) STEP 2 – Determine the cone half-apex angle, α, and compute the cone thickness, t_C, at the large end using paragraph 4.3.4.

c) STEP 3 – Proportion the transition geometry by assuming a value for the knuckle radius, r_k, and knuckle thickness, t_k, such that the following equations are satisfied. If all of these equations cannot be satisfied, then the cylinder-to-cone junction shall be designed in accordance with Part 5.

$$t_k \geq t_L \tag{4.3.63}$$

$$r_k > 3t_k \tag{4.3.64}$$

$$\frac{r_k}{R_L} > 0.03 \tag{4.3.65}$$

$$\alpha \le 60^\circ \tag{4.3.66}$$

d) STEP 4 – Determine the net section axial force, F_L, and bending moment , M_L, applied to the conical transition at the location of the knuckle. The thrust load due to pressure shall not be included as part of the axial force, F_L.

e) STEP 5 – Compute the stresses in the cylinder, knuckle, and cone at the junction using the equations in Table 4.3.7. The allowable stress criterion for a tensile stress is provided in Table 4.3.7. If either the hoop membrane stress, $\sigma_{\theta m}$, or axial membrane stress, σ_{sm}, at the junction is compressive, then the condition of local buckling shall be considered. Local buckling is not a concern if the limits given in Equations (4.3.67) and (4.3.68) are satisfied. F_{ha} is evaluated using paragraph 4.4.5.1, but substituting $F_{he} = 0.4E\left(\frac{t}{D_o}\right)$. F_{xa} is evaluated using paragraph 4.4.12.2.b with $\lambda = 0.15$. If the stresses of the acceptance criteria are satisfied, the design of the junction is complete.

$$\sigma_{\theta m} \le F_{ha} \tag{4.3.67}$$

$$\sigma_{sm} \le F_{xa} \tag{4.3.68}$$

f) STEP 6 – If the stress acceptance criterion in STEP 5 is satisfied, then the design is complete. If the stress acceptance criterion in STEP 5 is not satisfied, the knuckle thickness, cylinder thickness, or cone thickness near the junction may be increased until the stress acceptance criterion is satisfied. If the cylinder or cone thickness is increased, the section of increased thickness shall extend a length given by Equations (4.3.69) and (4.3.70), respectively. Proceed to STEP 3 to repeat the calculation with the new wall thicknesses.

$$L_{rcy} = K_m\sqrt{R_L t_L} \tag{4.3.69}$$

$$L_{rco} = K_m\sqrt{L_k t_C} \tag{4.3.70}$$

4.3.12.3 The procedure that shall be used to design the small end of a cylinder-to-cone junction with a flare is described below.

a) STEP 1 – Compute the small end cylinder thickness, t_S, using paragraph 4.3.3.

b) STEP 2 – Determine the cone half-apex angle, α, and compute the cone thickness, t_C, at the small end using paragraph 4.3.4.

c) STEP 3 – Proportion the transition geometry by assuming a value for the flare radius, r_f, and flare thickness, t_f, such that the following equations are satisfied. If all of these equations cannot be satisfied, then the cylinder-to-cone junction shall be designed in accordance with Part 5.

$$t_f \ge t_S \tag{4.3.71}$$

$$r_f > 3t_f \tag{4.3.72}$$

$$\frac{r_f}{R_S} > 0.03 \tag{4.3.73}$$

$$\alpha \leq 60° \tag{4.3.74}$$

d) STEP 4 – Determine the net section axial force, F_S, and bending moment, M_S, applied to the conical transition at the location of the knuckle. The thrust load due to pressure shall not be included as part of the axial force, F_S.

e) STEP 5 – Compute the stresses in the cylinder, flare, and cone at the junction using the equations in Table 4.3.8. The allowable stress criterion for a tensile stress is provided in Table 4.3.8. If either the hoop membrane stress, $\sigma_{\theta m}$, or axial membrane stress, σ_{sm}, at the junction is compressive, then the condition of local buckling shall be considered. Local buckling is not of concern if the limits given in Equation (4.3.67) and (4.3.68) are satisfied, using the procedure provided in paragraph 4.3.12.2.e. If the stresses of the acceptance criteria are satisfied, the design of the junction is complete.

f) STEP 6 – If the stress acceptance criterion in STEP 5 is satisfied, then the design is complete. If the stress acceptance criterion in STEP 5 is not satisfied, the knuckle thickness, cylinder thickness, or cone thickness near the junction may be increased until the stress acceptance criterion is satisfied. If the cylinder or cone thickness is increased, the section of increased thickness shall extend a length given by Equations (4.3.75) and (4.3.76), respectively. Proceed to STEP 3 to repeat the calculation with the new wall thicknesses.

$$L_{rcy} = K_m\sqrt{R_S t_S} \tag{4.3.75}$$

$$L_{rco} = K_m\sqrt{L_f t_C} \tag{4.3.76}$$

4.3.13 Nomenclature

A_R	cross-sectional area of the stiffening ring at the junction.
α	one-half of the apex angle of a conical shell.
α_1	cone angle in an offset transition.
α_2	cone angle in an offset transition.
B	curve-fit geometric constant
β_{co}	geometric factor for the cone.
β_{cy}	geometric factor for the cylinder.
β_f	angle used in the conical transition calculation when a flare is present.
β_{f1}	angle used in the conical transition calculation when a flare is present.
β_{f2}	angle used in the conical transition calculation when a flare is present.
β_k	angle used in the conical transition calculation when a knuckle is present.
β_{k1}	angle used in the conical transition calculation when a knuckle is present.
β_{k2}	angle used in the conical transition calculation when a knuckle is present.
β_{th}	angle used in the torispherical head calculation.

C_1	angle constant used in the torispherical head calculation.
C_2	angle constant used in the torispherical head calculation.
C_3	strength parameter used in the torispherical head calculation.
d	diameter of a drilled hole that does not completely penetrate a shell.
D	inside diameter of a shell or head.
D_o	outside diameter of a shell or head.
E_T	modulus of elasticity at maximum design temperature.
E_{RT}	modulus of elasticity at room temperature.
E	weld joint factor (see paragraph 4.2.4), the ligament efficiency (see paragraph 4.10.2), or the casting quality factor (see Part 3), as applicable, for the weld seam being evaluated (i.e. longitudinal or circumferential).
F	net-section axial force acting at the point of consideration, a positive force produces an axial tensile stress in the cylinder.
F_L	net-section axial force acting on the large end cylindrical shell, a positive force produces an axial tensile stress in the cylinder.
F_S	net-section axial force acting on the small end cylindrical shell, a positive force produces an axial tensile stress in the cylinder.
F_{ha}	allowable compressive hoop membrane stress as given in paragraph 4.4.
F_{xa}	allowable compressive axial membrane stress as given in paragraph 4.4.
G	constant used in the torispherical head calculation.
H	curve-fit geometric constant.
h	height of the ellipsoidal head measured to the inside surface.
I_R	moment of inertia of the stiffening ring at the junction.
j_k	number of locations around the knuckle that shall be evaluated, used in the conical transition stress calculation when a non-compact knuckle is present.
j_f	number of locations around the flare that shall be evaluated, used in the conical transition stress calculation when a non-compact flare is present.
k	angle constant used in the torispherical and elliptical head calculation.
K_m	length factor used in the conical transition calculation when a flare or knuckle is present.
K_{pc}	cylinder-to-cone junction plasticity correction factor.
λ	compressive stress factor.
L	inside crown radius of a torispherical head.
L_c	projected length of a conical shell.
L_f	length used in the conical transition stress calculation when a flare is present.
L_{1f}	length used in the conical transition stress calculation when a flare is present.
L_{1f}^j	length used in the conical transition stress calculation when a flare is present.
L_k	length used in the conical transition stress calculation when a knuckle is present.
L_{1k}	length used in the conical transition stress calculation when a knuckle is present.
L_{1k}^j	length used in the conical transition stress calculation when a knuckle is present.
M	net-section bending moment acting at the point of consideration.

M_{cs} total resultant meridional moment acting on the cone.

M_{csP} cylinder-to-cone junction resultant meridional moment acting on the cone, due to internal pressure.

M_{csX} cylinder-to-cone junction resultant meridional moment acting on the cone, due to an equivalent line load.

M_s total resultant meridional moment acting on the cylinder.

M_{sP} cylinder-to-cone junction resultant meridional moment acting on the cylinder, due to internal pressure.

M_{sX} cylinder-to-cone junction resultant meridional moment acting on the cylinder, due to an equivalent line load.

M_{sN} normalized curve-fit resultant meridional moment acting on the cylinder.

M_L net-section bending moment acting at the large end cylindrical shell.

M_S net-section bending moment acting at the small end cylindrical shell.

M_t net-section torsional moment acting on a shell section.

N_{cs} resultant meridional membrane force acting on the cone, due to pressure plus an equivalent line load.

$N_{c\theta}$ resultant circumferential membrane force acting on the cone, due to pressure plus an equivalent line load.

N_s resultant meridional membrane force acting on the cylinder, due to pressure plus an equivalent line load.

N_θ resultant circumferential membrane force acting on the cylinder, due to pressure plus an equivalent line load.

n ratio of the thickness of the cone to the thickness of the cylinder.

P internal design pressure.

P_a maximum allowable internal pressure of a torispherical head

P_{ac} allowable internal pressure of a torispherical head based on the rupture of the crown.

P_{ak} allowable internal pressure of a torispherical head based on a buckling failure of the knuckle.

P_{ck} value of internal pressure expected to result in a buckling failure of the knuckle in a torispherical head.

P_e equivalent design pressure used in the conical transition stress calculation when a knuckle or flare is present.

P_e^j equivalent design pressure at locations around the knuckle or flare, used in the conical transition stress calculation when a knuckle or flare is present.

P_{eth} value of internal pressure expected to produce elastic buckling of the knuckle in a torispherical head.

P_y value of the internal pressure expected to result in a maximum stress equal to the material yield strength in a torispherical head.

ϕ angle to locate a circumferential section in a spherical shell.

ϕ_f angle used in the conical transition calculation when a flare is present.

ϕ_f^j angle used in the conical transition calculation when a non-compact flare is present.

ϕ_f^e angle used in the conical transition calculation when a non-compact flare is present.

ϕ_f^s angle used in the conical transition calculation when a non-compact flare is present.

ϕ_k angle used in the conical transition calculation when a knuckle is present.

ϕ_k^j angle used in the conical transition calculation when a non-compact knuckle is present.

ϕ_k^e angle used in the conical transition calculation when a non-compact knuckle is present.

ϕ_k^s angle used in the conical transition calculation when a non-compact knuckle is present.

ϕ_{th} angle used in the torispherical head calculation.

Q total resultant shear force acting on the cylinder.

Q_c total resultant shear force acting on the cone.

Q_N normalized curve-fit resultant shear force acting on the cylinder.

Q_P cylinder-to-cone junction resultant shear force acting on the cylinder, due to internal pressure.

Q_X cylinder-to-cone junction resultant shear force acting on the cylinder, due to an equivalent line load.

R_C equivalent radius of the cone.

R_f radius to the center of curvature for the flare.

R_k radius to the center of curvature for the knuckle.

R_L inside radius of the large end of a conical transition.

R_m mean radius of the cylinder.

R_S inside radius of the small end of a conical transition.

R_{th} radius used in the torispherical head calculation.

r inside knuckle radius used in torispherical head calculation.

r_{hr} minimum hole radius.

r_k inside knuckle radius of the large end of a toriconical transition.

r_f inside flare radius of the small end of a toriconical transition.

s_R distance measured along the cylinder from the centroid of the stiffening ring centroid to the intersection of the cylinder and cone.

S allowable stress value from Annex 3.A evaluated at the design temperature.

S_a allowable stress amplitude.

S_{PS} allowable primary plus secondary stress evaluated using paragraph 5.5.6.1.d at the design temperature.

S_y yield strength from Annex 3.D evaluated at the design temperature.

σ_s axial (longitudinal) stress in a shell.

σ_{sm} axial (longitudinal) membrane stress in a shell.

σ_{sb} axial (longitudinal) bending stress in a shell.

σ_θ hoop (circumferential) stress in a shell.

$\sigma_{\theta m}$ hoop (circumferential) membrane stress in a shell.

$\sigma_{\theta b}$ hoop (circumferential) bending stress in a shell.

σ_1 principal stress in the 1-direction.

σ_2	principal stress in the 2-direction.
t	minimum required thickness of a shell.
t_C	thickness of the cone in a conical transition.
t_L	thickness of the large end cylinder in a conical transition.
t_S	thickness of the small end cylinder in a conical transition.
t_j	thickness of the cylinder, knuckle, or flue, as applicable, at the junction of a toriconical transition, $t_j \geq t$ and $t_j \geq t_c$.
t_{rw}	remaining wall thickness at the location of a partially drilled hole.
t_{rw1}	limit for the remaining wall thickness at the location of a partially drilled hole.
τ	torsional shear stress in a shell.
τ_{pd}	average shear stress in a shell at the location of a partially drilled hole. location where stress is computed for shells subject to supplemental loads. A value of zero defines the location of maximum positive longitudinal stress from net-section bending moment.
ν	Poisson's ratio.
X_L	equivalent line load acting on the large end cylinder, due to an axial force and bending moment.
X_S	equivalent line load acting on the small end cylinder, due to an axial force and bending moment.

4.3.14 Tables

Table 4.3.1 – Large End Junction

Cylinder	Cone
Stress Resultant Calculation	**Stress Resultant Calculation**
$M_{sP} = Pt_L^2 M_{sN}$, see Table 4.3.3	$M_{csP} = M_{sP}$
$M_{sX} = X_L t_L M_{sN}$, see Table 4.3.4	$M_{csX} = M_{sX}$
$M_s = M_{sP} + M_{sX}$	$M_{cs} = M_{csP} + M_{csX}$
$Q_P = Pt_L Q_N$, see Table 4.3.3	$Q_c = Q\cos[\alpha] + N_s \sin[\alpha]$ **(1)**
$Q_X = X_L Q_N$, see Table 4.3.4	$R_C = \dfrac{R_L}{\cos[\alpha]}$
$Q = Q_P + Q_X$	
$\beta_{cy} = \left[\dfrac{3(1-\nu^2)}{R_L^2 t_L^2}\right]^{0.25}$	$\beta_{co} = \left[\dfrac{3(1-\nu^2)}{R_C^2 t_C^2}\right]^{0.25}$
$N_s = \dfrac{PR_L}{2} + X_L$	$N_{cs} = N_s \cos[\alpha] - Q\sin[\alpha]$ **(2)**
$N_\theta = PR_L + 2\beta_{cy} R_L \left(-M_s \beta_{cy} + Q\right)$	$N_{c\theta} = \dfrac{PR_L}{\cos[\alpha]} + 2\beta_{co} R_C \left(-M_{cs}\beta_{co} - Q_c\right)$
$K_{pc} = 1.0$	$K_{cpc} = 1.0$
Stress Calculation	**Stress Calculation**
$\sigma_{sm} = \dfrac{N_s}{t_L}$	$\sigma_{sm} = \dfrac{N_{cs}}{t_C}$
$\sigma_{sb} = \dfrac{6M_s}{t_L^2 K_{pc}}$	$\sigma_{sb} = \dfrac{6M_{cs}}{t_C^2 K_{cpc}}$
$\sigma_{\theta m} = \dfrac{N_\theta}{t_L}$	$\sigma_{\theta m} = \dfrac{N_{c\theta}}{t_C}$
$\sigma_{\theta b} = \dfrac{6\nu M_s}{t_L^2 K_{pc}}$	$\sigma_{\theta b} = \dfrac{6\nu M_{cs}}{t_C^2 K_{cpc}}$
Acceptance Criteria	**Acceptance Criteria**
$\sigma_{sm} \le 1.5S$	$\sigma_{sm} \le 1.5S$
$\sigma_{sm} \pm \sigma_{sb} \le S_{PS}$	$\sigma_{sm} \pm \sigma_{sb} \le S_{PS}$
$\sigma_{\theta m} \le 1.5S$	$\sigma_{\theta m} \le 1.5S$
$\sigma_{\theta m} \pm \sigma_{\theta b} \le S_{PS}$	$\sigma_{\theta m} \pm \sigma_{\theta b} \le S_{PS}$

Notes:

1. The Q and N_s values used to determine the resultant shear force in the cone, Q_c, are the same as those defined for the cylinder.
2. The Q and N_s values used to determine the resultant meridional membrane force in the cone, N_{cs}, are the same as those defined for the cylinder.

Table 4.3.2 – Small End Junction

Cylinder	Cone
Stress Resultant Calculation	**Stress Resultant Calculation**
$M_{sP} = Pt_S^2 M_{sN}$, see Table 4.3.5	$M_{csP} = M_{sP}$
$M_{sX} = X_S t_S M_{sN}$, see Table 4.3.6	$M_{csX} = M_{sX}$
$M_s = M_{sP} + M_{sX}$	$M_{cs} = M_{csP} + M_{csX}$
$Q_P = Pt_S Q_N$, see Table 4.3.5	$Q_c = Q\cos\left[\alpha\right] + N_s \sin\left[\alpha\right]$ **(1)**
$Q_X = X_S Q_N$, see Table 4.3.6	$R_C = \dfrac{R_S}{\cos\left[\alpha\right]}$
$Q = Q_P + Q_X$	
$\beta_{cy} = \left[\dfrac{3(1-\nu^2)}{R_S^2 t_S^2}\right]^{0.25}$	$\beta_{co} = \left[\dfrac{3(1-\nu^2)}{R_C^2 t_C^2}\right]^{0.25}$
$N_s = \dfrac{PR_S}{2} + X_S$	$N_{cs} = N_s \cos\left[\alpha\right] - Q\sin\left[\alpha\right]$ **(2)**
$N_\theta = PR_S + 2\beta_{cy} R_S \left(-M_s \beta_{cy} - Q\right)$	$N_{c\theta} = \dfrac{PR_S}{\cos\left[\alpha\right]} + 2\beta_{co} R_C \left(-M_{cs}\beta_{co} + Q_c\right)$
$K_{pc} = 1.0$	$K_{cpc} = 1.0$
Stress Calculation	**Stress Calculation**
$\sigma_{sm} = \dfrac{N_s}{t_S}$	$\sigma_{sm} = \dfrac{N_{cs}}{t_C}$
$\sigma_{sb} = \dfrac{6M_s}{t_S^2 K_{pc}}$	$\sigma_{sb} = \dfrac{6M_{cs}}{t_C^2 K_{cpc}}$
$\sigma_{\theta m} = \dfrac{N_\theta}{t_S}$	$\sigma_{\theta m} = \dfrac{N_{c\theta}}{t_C}$
$\sigma_{\theta b} = \dfrac{6\nu M_s}{t_S^2 K_{pc}}$	$\sigma_{\theta b} = \dfrac{6\nu M_{cs}}{t_C^2 K_{cpc}}$
Acceptance Criteria	**Acceptance Criteria**
$\sigma_{sm} \le 1.5S$	$\sigma_{sm} \le 1.5S$
$\sigma_{sm} \pm \sigma_{sb} \le S_{PS}$	$\sigma_{sm} \pm \sigma_{sb} \le S_{PS}$
$\sigma_{\theta m} \le 1.5S$	$\sigma_{\theta m} \le 1.5S$
$\sigma_{\theta m} \pm \sigma_{\theta b} \le S_{PS}$	$\sigma_{\theta m} \pm \sigma_{\theta b} \le S_{PS}$

Notes:

1. The Q and N_s values used to determine the resultant shear force in the cone, Q_c , are the same as those defined for the cylinder.
2. The Q and N_s values used to determine the resultant meridional membrane force in the cone, N_{cs} , are the same as those defined for the cylinder.

Table 4.3.3 – Pressure Applied To Large End Junction

Junction Moment Resultant – M_{sN} (1)					
Equation Coefficients – C_i	n				
	1	1.25	1.5	1.75	2
1	-3.065534	-3.113501	-3.140885	-3.129850	-3.115764
2	3.642747	3.708036	3.720338	3.674582	3.623956
3	0.810048	0.736679	0.623373	0.490738	0.360998
4	-0.221192	-0.239151	-0.241393	-0.224678	-0.209963
5	-0.081824	-0.075734	-0.056744	-0.034581	-0.013613
6	0.035052	0.083171	0.157222	0.240314	0.316184
7	0.025775	0.027432	0.027393	0.025163	0.023508
8	-0.015413	-0.015659	-0.017311	-0.019456	-0.021796
9	0.002102	0.000993	-0.004600	-0.011145	-0.017172
10	-0.005587	-0.013283	-0.025609	-0.039144	-0.050859
Junction Shear Force Resultant – Q_N (1)					
1	-1.983852	-1.911375	-1.893640	-1.852083	-1.816642
2	2.410703	2.292069	2.253430	2.184549	2.126469
3	0.626443	0.478030	0.364794	0.251818	0.152468
4	-0.119151	-0.079165	-0.075123	-0.059024	-0.048876
5	-0.115841	-0.074658	-0.047032	-0.024214	-0.007486
6	0.122993	0.219247	0.282565	0.343492	0.390839
7	0.012160	0.007250	0.007505	0.006116	0.005632
8	-0.016987	-0.021607	-0.024667	-0.027144	-0.029118
9	0.010919	-0.003818	-0.012439	-0.018971	-0.023076
10	-0.016653	-0.033814	-0.043500	-0.052435	-0.058417

Note: (1) The equation to determine M_{sN} and Q_N is shown below.

$$M_{sN}, Q_N = -\exp\left[\begin{array}{l} C_1 + C_2\ln[H] + C_3\ln[B] + C_4\left(\ln[H]\right)^2 + C_5\left(\ln[B]\right)^2 + C_6\ln[H]\ln[B] + \\ C_7\left(\ln[H]\right)^3 + C_8\left(\ln[B]\right)^3 + C_9\ln[H]\left(\ln[B]\right)^2 + C_{10}\left(\ln[H]\right)^2\ln[B] \end{array}\right]$$

Table 4.3.4 – Equivalent Line Load Applied To Large End Junction

Junction Moment Resultant – M_{sN} (1)					
Equation Coefficients – C_i	n				
	1	**1.25**	**1.5**	**1.75**	**2**
1	-5.697151	-5.727483	-5.893323	-6.159334	-6.532748
2	0.003838	0.006762	0.012440	0.019888	0.029927
3	0.476317	0.471833	0.466370	0.461308	0.454550
4	-0.213157	-0.213004	-0.211065	-0.207037	-0.200411
5	2.233703	2.258541	2.335015	2.449057	2.606550
6	0.000032	0.000010	-0.000006	-0.000008	-0.000004
7	0.002506	0.003358	0.004949	0.007005	0.009792
8	-0.001663	-0.002079	-0.003105	-0.004687	-0.007017
9	-0.212965	-0.216613	-0.224714	-0.235979	-0.251220
10	0.000138	-0.000108	-0.000721	-0.001597	-0.002797
11	-0.106203	-0.106269	-0.107142	-0.108733	-0.110901
Junction Shear Force Resultant – Q_N (1)					
1	-4.774616	-5.125169	-5.556823	-6.113380	-6.858200
2	0.000461	0.021875	0.049082	0.084130	0.131374
3	-0.002831	-0.055928	-0.127941	-0.225294	-0.361885
4	-0.197117	-0.196848	-0.196204	-0.194732	-0.193588
5	1.982132	2.156708	2.378102	2.668633	3.069269
6	0.000069	-0.000450	-0.001077	-0.001821	-0.002760
7	-0.000234	0.000188	0.000821	0.001694	0.002958
8	-0.003536	-0.005341	-0.007738	-0.010934	-0.015089
9	-0.202493	-0.223872	-0.251223	-0.287283	-0.337767
10	-0.000088	-0.002426	-0.005428	-0.009440	-0.015045
11	0.001365	0.012698	0.027686	0.047652	0.075289

Notes: (1) The equation to determine M_{sN} and Q_N is shown below.

$$M_{sN},\, Q_N = -\exp\left[\frac{C_1+C_3\ln\left[H^2\right]+C_5\ln\left[\alpha\right]+C_7\left(\ln\left[H^2\right]\right)^2+C_9\left(\ln\left[\alpha\right]\right)^2+C_{11}\ln\left[H^2\right]\ln\left[\alpha\right]}{1+C_2\ln\left[H^2\right]+C_4\ln\left[\alpha\right]+C_6\left(\ln\left[H^2\right]\right)^2+C_8\left(\ln\left[\alpha\right]\right)^2+C_{10}\ln\left[H^2\right]\ln\left[\alpha\right]}\right]$$

Table 4.3.5 – Pressure Applied To Small End Junction

Equation Coefficients – C_i	n = 1	1.25	1.5	1.75	2
Junction Moment Resultant – M_{sN} (1)					
1	-9.615764	-10.115298	-11.531005	-14.040332	-18.457734
2	1.755095	1.858053	2.170806	2.762452	3.859890
3	3.937841	4.222547	4.872664	5.973215	7.923210
4	-0.043572	-0.053476	-0.080011	-0.131830	-0.228146
5	-1.035596	-1.100505	-1.213287	-1.388782	-1.685101
6	-0.008908	-0.033941	-0.121942	-0.288589	-0.612009
7	0.003984	0.004388	0.005287	0.006975	0.010041
8	0.115270	0.121595	0.129218	0.139465	0.154368
9	0.013712	0.015269	0.022097	0.034632	0.059879
10	-0.007031	-0.006067	-0.002848	0.003867	0.017109
Junction Shear Force Resultant – Q_N (2)					
1	0.028350	0.207327	0.376538	0.532382	0.682418
2	0.000020	0.000007	-0.000008	-0.000023	-0.000040
3	0.001668	0.003856	0.005918	0.007947	0.009881
4	0.002987	0.002885	0.002781	0.002709	0.002632
5	0.001125	-0.000330	-0.001848	-0.002664	-0.003542
6	0.000000	0.000000	0.000000	0.000000	0.000000
7	0.000001	-0.000001	-0.000003	-0.000005	-0.000006
8	-0.000122	-0.000120	-0.000118	-0.000117	-0.000116
9	-0.000181	-0.000139	-0.000106	-0.000090	-0.000079
10	0.000001	0.000001	0.000001	0.000001	0.000001
11	-0.004724	-0.004417	-0.004128	-0.003847	-0.003570

Notes:

1. The equation to determine M_{sN} is shown below.

$$M_{sN} = \exp\left[\begin{array}{l} C_1 + C_2\ln\left[H^2\right] + C_3\ln\left[\alpha\right] + C_4\left(\ln\left[H^2\right]\right)^2 + C_5\left(\ln\left[\alpha\right]\right)^2 + C_6\ln\left[H^2\right]\ln\left[\alpha\right] + \\ C_7\left(\ln\left[H^2\right]\right)^3 + C_8\left(\ln\left[\alpha\right]\right)^3 + C_9\ln\left[H^2\right]\left(\ln\left[\alpha\right]\right)^2 + C_{10}\left(\ln\left[H^2\right]\right)^2\ln\left[\alpha\right] \end{array}\right]$$

2. The equation to determine Q_N is shown below.

$$Q_N = \left(\frac{C_1 + C_3H^2 + C_5\alpha + C_7H^4 + C_9\alpha^2 + C_{11}H^2\alpha}{1 + C_2H^2 + C_4\alpha + C_6H^4 + C_8\alpha^2 + C_{10}H^2\alpha}\right)$$

Table 4.3.6 – Equivalent Line Load Applied To Small End Junction

Equation Coefficients – C_i	n = 1	1.25	1.5	1.75	2
Junction Moment Resultant – M_{sN} (1)					
1	-0.000792	0.000042	0.002412	0.005766	0.009868
2	-0.000627	-0.000327	-0.000033	0.000236	0.000453
3	-0.001222	-0.001188	-0.001079	-0.000951	-0.000860
4	0.142039	0.132463	0.125812	0.121877	0.120814
5	0.010704	0.009735	0.009802	0.010465	0.010928
6	0.000013	0.000006	-0.000002	-0.000009	-0.000015
7	-0.000006	-0.000001	-0.000006	-0.000008	-0.000008
8	0.009674	0.008839	0.007580	0.006261	0.005044
9	0.006254	0.005493	0.003701	0.001619	0.000381
10	-0.000046	0.000011	0.000088	0.000171	0.000230
11	0.202195	0.208304	0.205169	0.197061	0.186547
Junction Shear Force Resultant – Q_N (2)					
1	-0.460579	-0.444768	-0.428659	-0.412043	-0.396046
2	-0.002381	0.006711	0.013388	0.019509	0.026272
3	-0.400925	-0.376106	-0.353464	-0.331009	-0.309046
4	0.001550	-0.000672	-0.002169	-0.003562	-0.005266
5	-0.140077	-0.129459	-0.121074	-0.113195	-0.105461
6	0.000793	0.001950	0.002212	0.002168	0.002310
7	-0.000219	-0.000023	0.000098	0.000215	0.000374
8	-0.019081	-0.017115	-0.015814	-0.014699	-0.013625
9	0.000384	0.000618	0.000739	0.000806	0.000860
10	0.000103	0.000006	0.000038	0.000102	0.000117

Notes:

1. The equation to determine M_{sN} is shown below.

$$M_{sN} = \left(\frac{C_1 + C_3 H + C_5 B + C_7 H^2 + C_9 B^2 + C_{11} HB}{1 + C_2 H + C_4 B + C_6 H^2 + C_8 B^2 + C_{10} HB} \right)$$

2. The equation to determine Q_N is shown below.

$$Q_N = \left(\begin{array}{l} C_1 + C_2 \ln[H] + C_3 \ln[B] + C_4 (\ln[H])^2 + C_5 (\ln[B])^2 + C_6 \ln[H]\ln[B] + \\ C_7 (\ln[H])^3 + C_8 (\ln[B])^3 + C_9 \ln[H](\ln[B])^2 + C_{10} (\ln[H])^2 \ln[B] \end{array} \right)$$

Table 4.3.7 – Stress Calculations – Knuckle – Large End Cylinder

Compact Knuckle: $\alpha r_k < 2K_m \left(\left\{ R_k \left(\alpha^{-1} \cdot \tan[\alpha] \right)^{0.5} + r_k \right\} t_k \right)^{0.5} \qquad where \;\; K_m = 0.7$

Stress Calculation

$$\sigma_{\theta m} = \frac{PK_m \left(R_L \sqrt{R_L t_L} + L_k \sqrt{L_k t_C} \right) + \alpha \left(PL_{1k} r_k - 0.5 P_e L_{1k}^2 \right)}{K_m \left(t_L \sqrt{R_L t_L} + t_C \sqrt{L_k t_C} \right) + \alpha t_k r_k}$$

$$\sigma_{sm} = \frac{P_e L_{1k}}{2t_k}$$

$$P_e = P + \frac{F_L}{\pi L_{1k}^2 \cos^2\left[\dfrac{\alpha}{2}\right]} \pm \frac{2M_L}{\pi L_{1k}^3 \cos^3\left[\dfrac{\alpha}{2}\right]}$$

$$L_k = \frac{R_k}{\cos[\alpha]} + r_k$$

$$L_{1k} = R_k \left(\alpha^{-1} \cdot \tan[\alpha] \right)^{0.5} + r_k$$

Acceptance Criteria

$\sigma_{\theta m} \le S \;\; \sigma_{sm} \le S$

Non-Compact Knuckle: $\alpha r_k \ge 2K_m \left(\left\{ R_k \left(\alpha^{-1} \cdot \tan[\alpha] \right)^{0.5} + r_k \right\} t_k \right)^{0.5} \qquad where \;\; K_m = 0.7$

Stress Calculation At TL-1

$$\sigma_{\theta m} = \frac{PR_L K_m \sqrt{R_L t_L} + \beta_k \left(PL_{1k} r_k - 0.5 P_e L_{1k}^2 \right)}{K_m \left(t_L \sqrt{R_L t_L} + t_k \sqrt{L_{1k} t_k} \right)}$$

$$\sigma_{sm} = \frac{P_e L_{1k}}{2t_k}$$

$$P_e = P + \frac{F_L}{\pi L_{1k}^2 \cos^2\left[\dfrac{\beta_k}{2}\right]} \pm \frac{2M_L}{\pi L_{1k}^3 \cos^3\left[\dfrac{\beta_k}{2}\right]}$$

$$L_{1k} = R_k \left(\beta_k^{-1} \cdot \tan[\beta_k] \right)^{0.5} + r_k$$

$$\beta_k = \left(\frac{K_m}{r_k} \right) \sqrt{R_L t_k}$$

Stress Calculation At TL-2

$$\sigma_{\theta m} = \frac{PL_k K_m \sqrt{L_k t_C} + (\alpha - \beta_k)\left(PL_{1k} r_k - 0.5 P_e L_{1k}^2 \right)}{K_m \left(t_C \sqrt{L_k t_C} + t_k \sqrt{L_{1k} t_k} \right)}$$

$$\sigma_{sm} = \frac{P_e L_{1k}}{2t_k}$$

$$P_e = P + \frac{F_L}{\pi L_{1k}^2 \cos^2[\phi_k]} \pm \frac{2M_L}{\pi L_{1k}^3 \cos^3[\phi_k]}$$

$$L_k = \frac{R_k}{\cos[\alpha]} + r_k$$

$$L_{1k} = R_k \left(\left\{ \tan[\alpha] - \tan[\beta_k] \right\} \left\{ \alpha - \beta_k \right\}^{-1} \right)^{0.5} + r_k$$

$$\beta_k = \alpha - \left(\frac{K_m}{r_k} \right) \sqrt{L_k t_k}$$

$$\phi_k = \frac{(\alpha + \beta_k)}{2}$$

Table 4.3.7 – Stress Calculations – Knuckle – Large End Cylinder

Stress Calculation In The Non-Compact Knuckle Region

Note: The number of locations around the knuckle that shall be evaluated is given by the following equation.

$$j_k = 2\left(\text{int}\left[\frac{\alpha - \beta_{k1} - \beta_{k2}}{\beta_{k1} + \beta_{k2}} \right] + 1 \right) + 1$$

where

$$\beta_{k1} = \frac{K_m}{r_k}\sqrt{R_L t_k}$$

$$\beta_{k2} = \frac{K_m}{r_k}\sqrt{L_k t_k}$$

$$L_k = \frac{R_k}{\cos[\alpha]} + r_k$$

For $j = 1, ..., j_k$, *compute*

$$\sigma_{0m}^j = \frac{PL_{1k}^j}{t_k} - \frac{P_e^j \left(L_{1k}^j\right)^2}{2r_k t_k}$$

$$\sigma_{sm}^j = \frac{P_e^j L_{1k}^j}{2t_k}$$

where

$$P_e^j = P + \frac{F_L}{\pi \left(L_{1k}^j\right)^2 \cos^2\left[\phi_k^j\right]} \pm \frac{2M_L}{\pi \left(L_{1k}^j\right)^3 \cos^3\left[\phi_k^j\right]}$$

$$L_{1k}^j = \frac{R_k}{\cos\left[\phi_k^j\right]} + r_k$$

$$\phi_k^j = \phi_k^s + (j-1)\left(\frac{\phi_k^e - \phi_k^s}{j_k - 1} \right)$$

$$\phi_k^s = \beta_{k1}$$

$$\phi_k^e = \alpha - \beta_{k2}$$

Acceptance Criteria

$$\sigma_{\theta m} \le S \;\; \sigma_{sm} \le S$$

$$\sigma_{\theta m}^j \le S \;\; \sigma_{sm}^j \le S$$

Table 4.3.8 – Stress Calculations – Flare – Small End Cylinder

Compact Flare: $\alpha r_f < 2K_m \left(\left\{ R_f \left(\alpha^{-1} \cdot \tan[\alpha] \right)^{0.5} - r_f \right\} t_f \right)^{0.5} \quad where \;\; K_m = 0.7$

Stress Calculation

$$\sigma_{\theta m} = \frac{PK_m \left(R_S \sqrt{R_S t_S} + L_f \sqrt{L_f t_C} \right) + \alpha \left(PL_{1f} r_f + 0.5 P_e L_{1f}^2 \right)}{K_m \left(t_S \sqrt{R_S t_S} + t_C \sqrt{L_f t_C} \right) + \alpha t_f r_f}$$

$$\sigma_{sm} = \frac{P_e L_{1f}}{2t_f}$$

$$P_e = P + \frac{F_S}{\pi L_{1f}^2 \cos^2\left[\frac{\alpha}{2}\right]} \pm \frac{2M_S}{\pi L_{1f}^3 \cos^3\left[\frac{\alpha}{2}\right]}$$

$$L_f = \frac{R_f}{\cos[\alpha]} - r_f$$

$$L_{1f} = R_f \left(\alpha^{-1} \cdot \tan[\alpha] \right)^{0.5} - r_f$$

Acceptance Criteria

$\sigma_{\theta m} \le S \;\; \sigma_{sm} \le S$

Non-Compact Flare: $\alpha r_f \ge 2K_m \left(\left\{ R_f \left(\alpha^{-1} \cdot \tan[\alpha] \right)^{0.5} - r_f \right\} t_f \right)^{0.5} \quad where \;\; K_m = 0.7$

Stress Calculation At TL-3

$$\sigma_{\theta m} = \frac{PL_f K_m \sqrt{L_f t_C} + \left(\alpha - \beta_f \right) \left(PL_{1f} r_f + 0.5 P_e \right.}{K_m \left(t_C \sqrt{L_f t_C} + t_f \sqrt{L_{1f} t_f} \right)}$$

$$\sigma_{sm} = \frac{P_e L_{1f}}{2t_f}$$

$$P_e = P + \frac{F_S}{\pi L_{1f}^2 \cos^2\left[\phi_f\right]} \pm \frac{2M_S}{\pi L_{1f}^3 \cos^3\left[\phi_f\right]}$$

$$L_f = \frac{R_f}{\cos[\alpha]} - r_f$$

$$L_{1f} = R_f \left(\left\{ \tan[\alpha] - \tan\left[\beta_f\right] \right\} \left\{ \alpha - \beta_f \right\}^{-1} \right)^{0.5} - r$$

$$\beta_f = \alpha - \left(\frac{K_m}{r_f} \right) \sqrt{L_f t_f}$$

$$\phi_f = \frac{\left(\alpha + \beta_f \right)}{2}$$

Stress Calculation At TL-4

$$\sigma_{\theta m} = \frac{PR_S K_m \sqrt{R_S t_S} + \beta_f \left(PL_{1f} r_f + 0.5 P_e L_{1f}^2 \right)}{K_m \left(t_S \sqrt{R_S t_S} + t_f \sqrt{L_{1f} t_f} \right)}$$

$$\sigma_{sm} = \frac{P_e L_{1f}}{2t_f}$$

$$P_e = P + \frac{F_S}{\pi L_{1f}^2 \cos^2\left[\frac{\beta_f}{2}\right]} \pm \frac{2M_S}{\pi L_{1f}^3 \cos^3\left[\frac{\beta_f}{2}\right]}$$

$$L_{1f} = R_f \left(\beta_f^{-1} \cdot \tan\left[\beta_f\right] \right)^{0.5} - r_f \quad \beta_f = \left(\frac{K_m}{r_f} \right) \sqrt{R_S t_f}$$

Table 4.3.8 – Stress Calculations – Flare – Small End Cylinder

Stress Calculation In The Non-Compact Flare Region

Note: The number of locations around the flare that shall be evaluated is given by the following equation.

$$j_f = 2\left(\text{int}\left[\frac{\alpha - \beta_{f1} - \beta_{f2}}{\beta_{f1} + \beta_{f2}} \right] + 1 \right) + 1$$

where

$$\beta_{f1} = \frac{K_m}{r_f}\sqrt{L_f t_f}$$

$$\beta_{f2} = \frac{K_m}{r_f}\sqrt{R_s t_f}$$

$$L_f = \frac{R_f}{\cos[\alpha]} - r_f$$

For $j = 1, \ldots, j_f$, *compute*

$$\sigma_{\theta m}^j = \frac{P L_{1f}^j}{t_f} + \frac{P_e^j \left(L_{1f}^j\right)^2}{2 r_f t_f}$$

$$\sigma_{sm}^j = \frac{P_e^j L_{1f}^j}{2 t_f}$$

where

$$P_e^j = P + \frac{F_S}{\pi \left(L_{1f}^j\right)^2 \cos^2\left[\phi_f^j\right]} \pm \frac{2 M_S}{\pi \left(L_{1f}^j\right)^3 \cos^3\left[\phi_f^j\right]}$$

$$L_{1f}^j = \frac{R_f}{\cos\left[\phi_f^j\right]} - r_f$$

$$\phi_f^j = \phi_f^s - (j-1)\left(\frac{\phi_f^s - \phi_f^e}{j_f - 1} \right)$$

$$\phi_f^s = \alpha - \beta_{f1}$$

$$\phi_f^e = \beta_{f2}$$

Acceptance Criteria

$\sigma_{\theta m} \le S \quad \sigma_{sm} \le S$

$\sigma_{\theta m}^j \le S \quad \sigma_{sm}^j \le S$

4.3.15 Figures

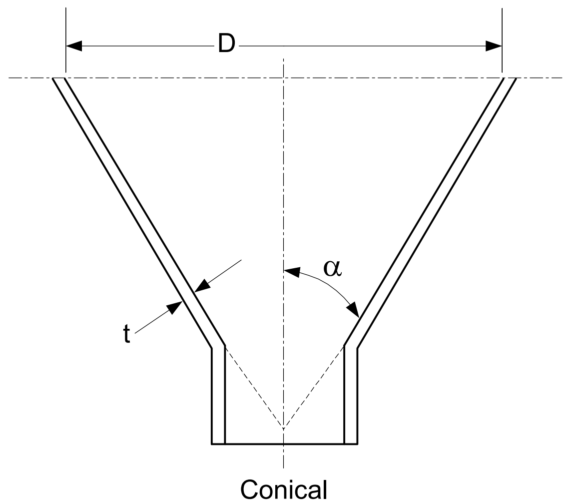


Figure 4.3.1 – Conical Shell

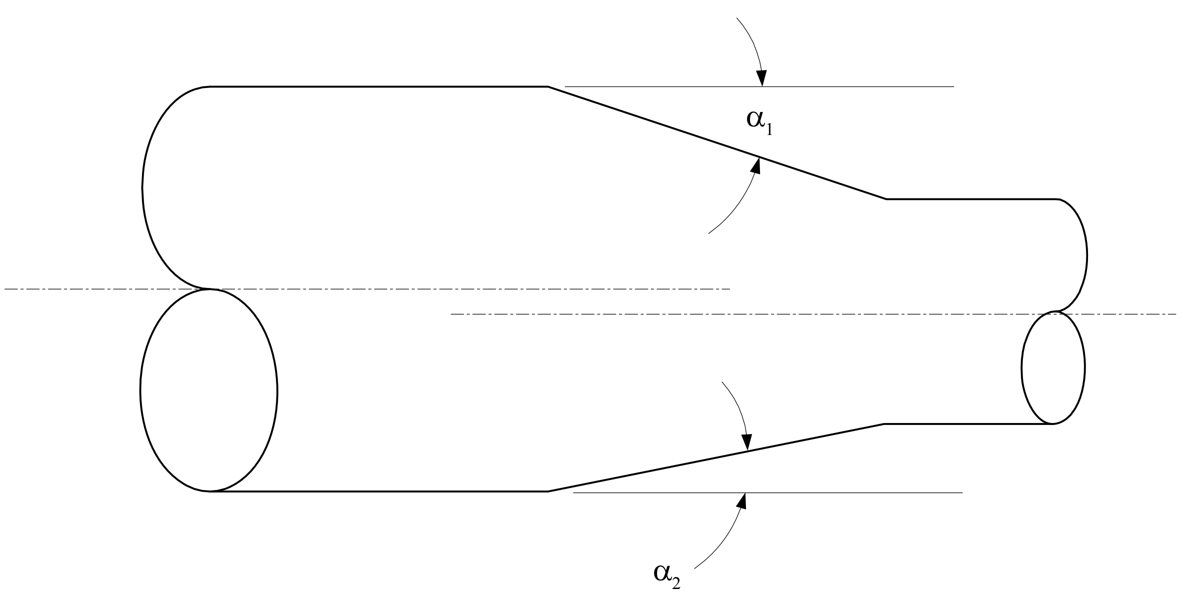


Figure 4.3.2 – Offset Transition Detail

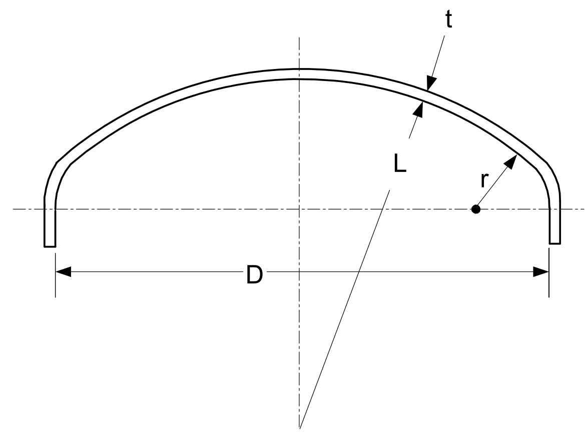


Figure 4.3.3 – Torispherical Head of Uniform Thickness

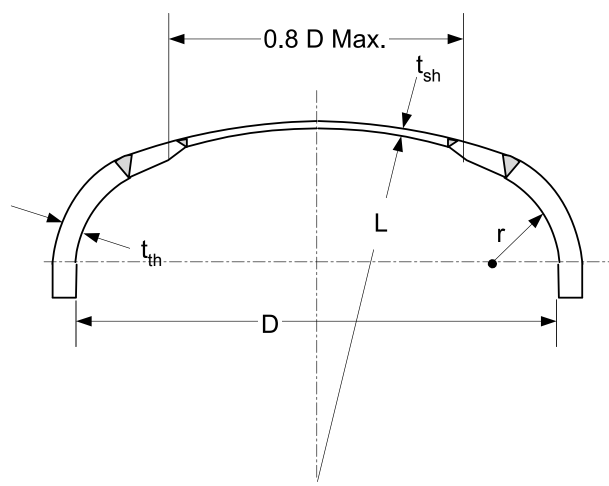


Figure 4.3.4 – Torispherical Head of Different Thickness of Dome and Knuckle

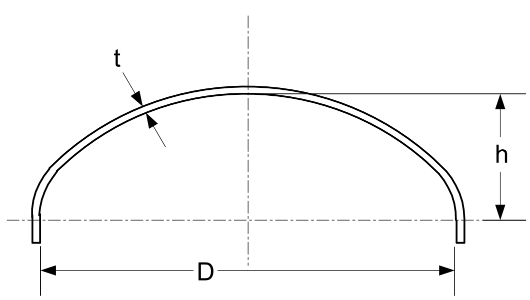


Figure 4.3.5 – Ellipsoidal Head

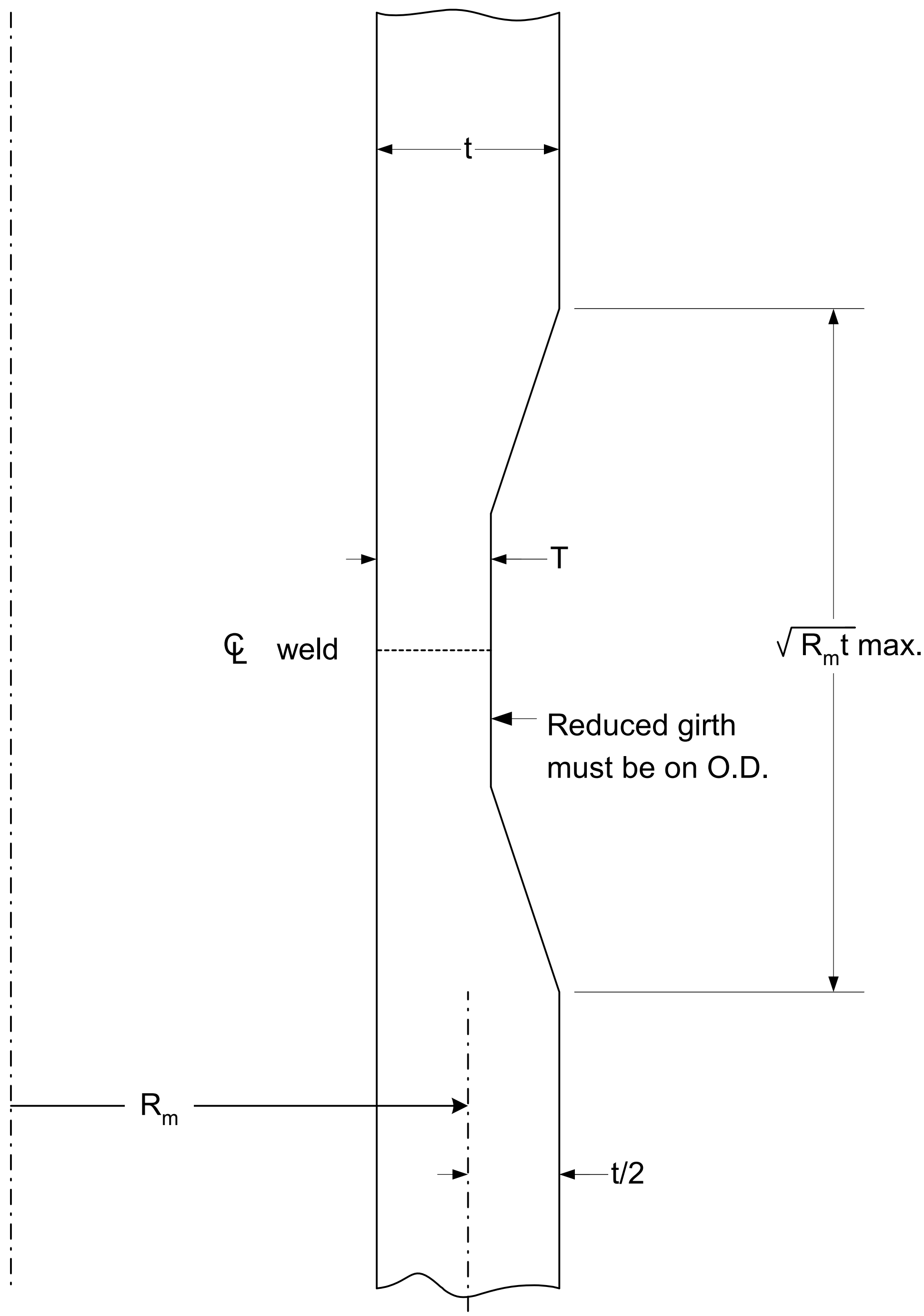
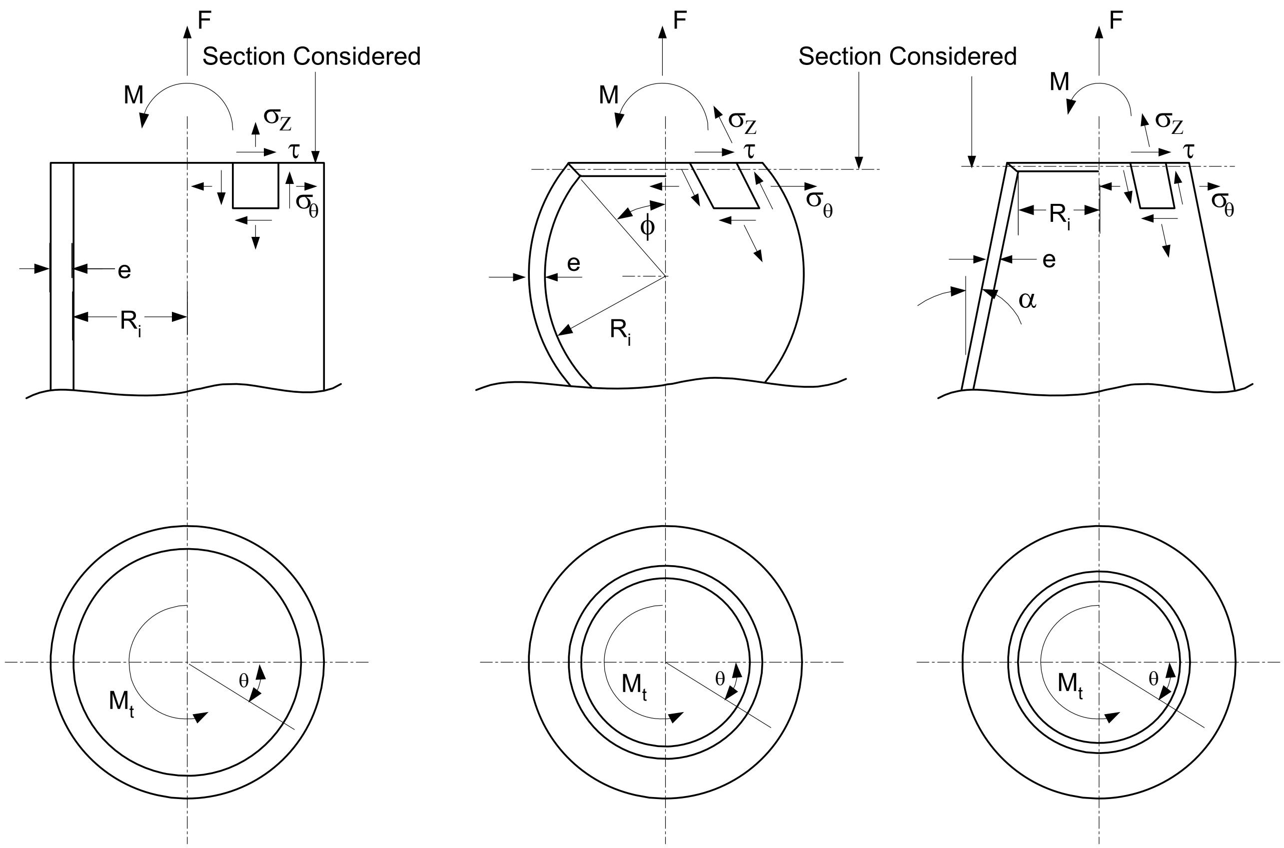


Figure 4.3.6 – Local Thin Band in a Cylindrical Shell

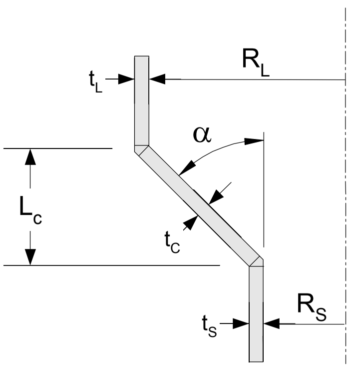


(a) Stresses Within A Cylindrical Shell

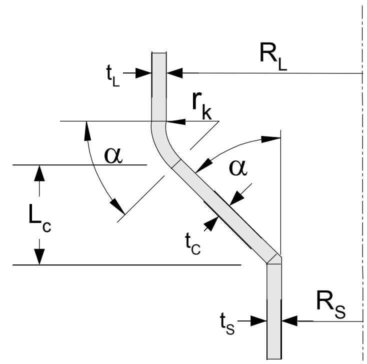
(b) Stresses Within A Spherical Shell

(c) Stresses Within A Conical Shell

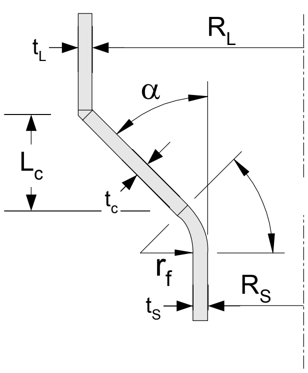
Figure 4.3.7 – Shells Subjected to Supplemental Loadings



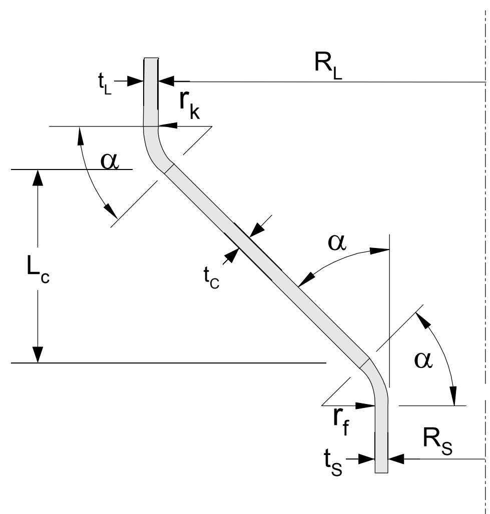
(a) Cone without a Knuckle at Large and without a Flare at the Small End



(b) Cone with a Knuckle at Large and without a Flare at the Small End



(c) Cone without a Knuckle at Large and with a Flare at the Small End



(d) Cone with a Knuckle at Large and with a Flare at the Small End

Figure 4.3.8 – Conical Transition Details

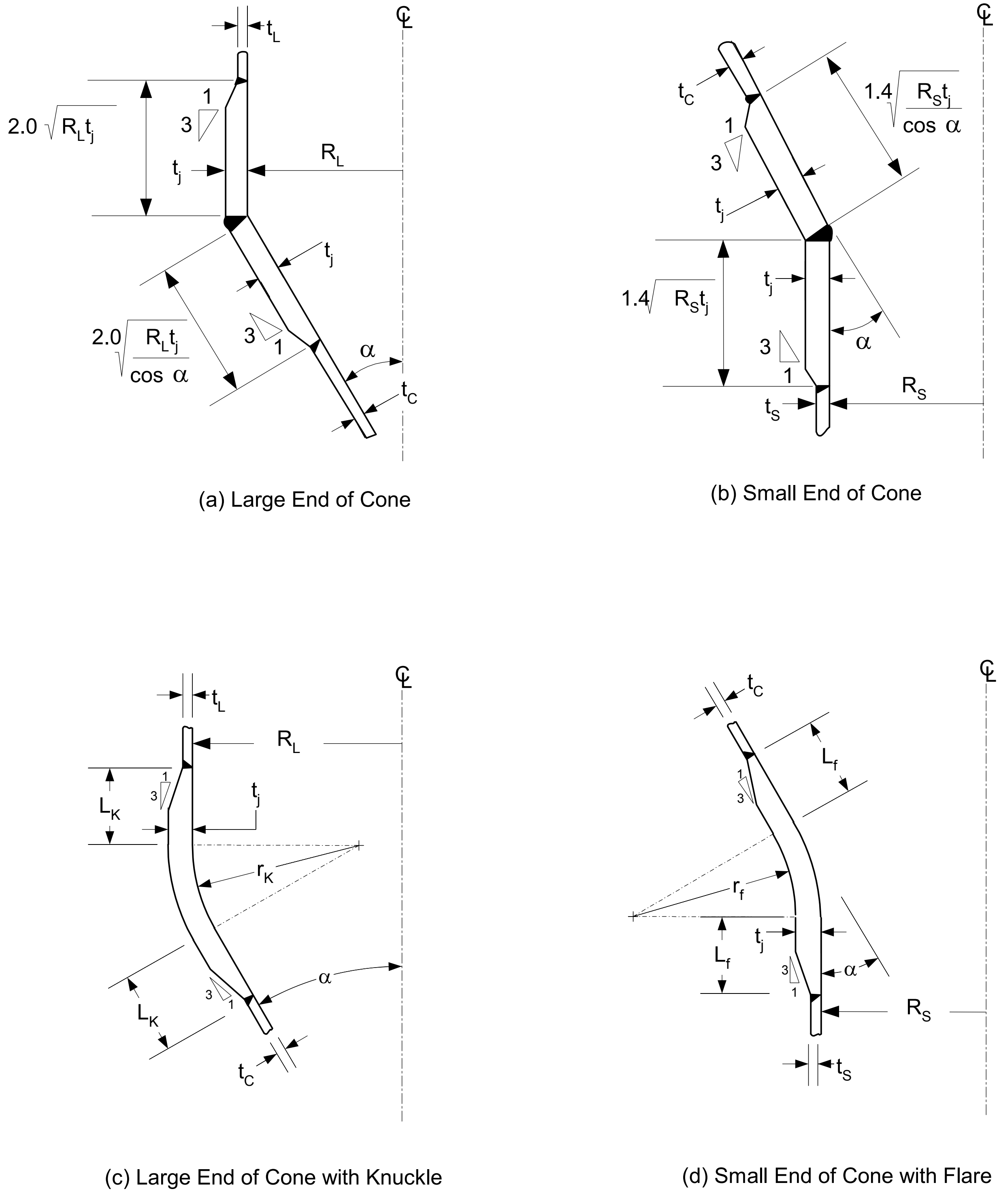


Figure 4.3.9 – Reinforcement Requirements for Conical Transition Junction

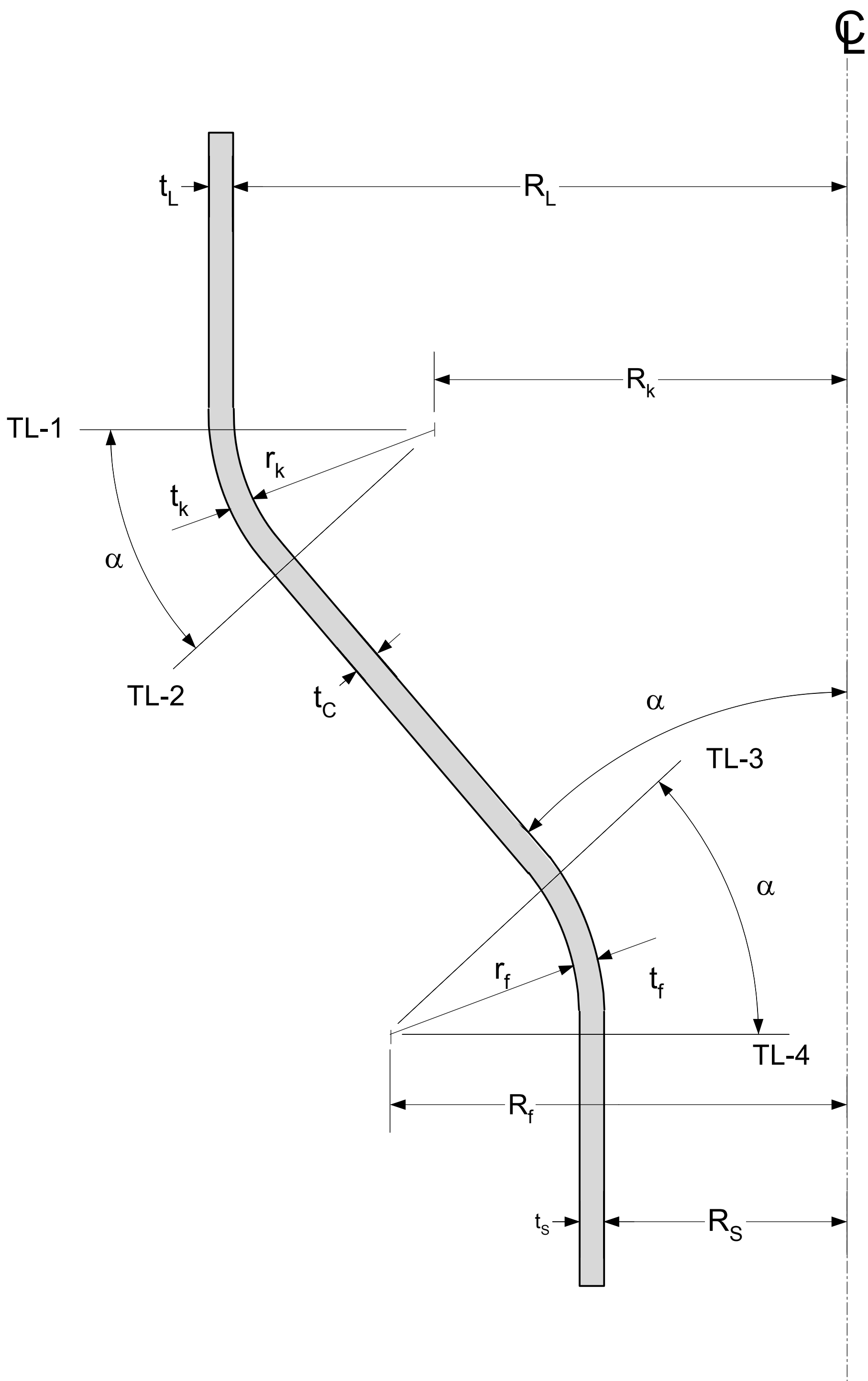


Figure 4.3.10 – Parameters for Knuckle and Flare Design

4.4 Design of Shells Under External Pressure and Allowable Compressive Stresses

4.4.1 Scope

4.4.1.1 Paragraph 4.4 provides rules for determining the required wall thickness of cylindrical, conical, spherical, torispherical, and ellipsoidal shells and heads subject to external pressure. In this context, external pressure is defined as pressure acting on the convex side of the shell.

4.4.1.2 The effects of supplemental loads are not included in the design equations for shells and heads included in paragraphs 4.4.5 to 4.4.9. The effects of supplemental loads that result in combined loadings shall be evaluated in a separate analysis performed in accordance with the methods in paragraph 4.4.12.

4.4.1.3 Rules are also provided for the design of cylindrical-to-conical shell transition junctions in paragraphs 4.4.13 and 4.4.14. To facilitate the use of these rules, it is recommended that the shell wall thickness and stiffener configuration, as applicable, first be designed using the rules in paragraphs 4.4.5 through 4.4.9. After an initial design is determined, this design should then be checked and modified as required using the rules of paragraphs 4.4.13 and 4.4.14.

4.4.1.4 The equations in this paragraph are applicable for $D_o/t \le 2000$. If $D_o/t > 2000$, then the design shall be in accordance with Part 5. In developing the equations in the paragraph, the shell section is assumed to be axisymmetric with uniform thickness for unstiffened cylinders and formed heads. Stiffened cylinders and cones are also assumed to be of uniform thickness between stiffeners. Where nozzles with reinforcing plates or locally thickened shell sections exist, the thinnest uniform thickness in the applicable unstiffened or stiffened shell section shall be used for the calculation of the allowable compressive stress.

4.4.1.5 Special consideration shall be given to ends of components (shell sections) or areas of load application where stress distribution may be in the inelastic range and localized stresses may exceed those predicted by linear theory.

4.4.1.6 When the localized stresses extend over a distance equal to one-half the length of the buckling mode (approximately $1.2\sqrt{D_o t}$), the localized stresses shall be considered as a uniform stress for the design of the shell section.

4.4.1.7 The buckling strength formulations presented in this paragraph are based upon linear structural stability theory which is modified by reduction factors which account for the effects of imperfections, boundary conditions, non-linearity of material properties and residual stresses. The reduction factors are determined from approximate lower bound values of test data of shells with initial imperfections representative of the tolerance limits specified in this paragraph.

4.4.2 Design Factors

The allowable stresses are determined by applying a design factor, FS, to the predicted buckling stresses. The required values of FS are 2.0 when the buckling stress is elastic and 1.667 when the predicted buckling stress equals the minimum specified yield strength at the design temperature. A linear variation shall be used between these limits. The equations for FS are given below where F_{ic} is the predicted buckling stress that is determined by setting $FS = 1.0$ in the allowable stress equations. For combinations of design loads and earthquake loading or wind loading (see paragraph 4.1.5.3), the allowable stress for F_{bha} or F_{ba} in Equations (4.4.106), (4.4.107), (4.4.108), (4.4.111), (4.4.112) and (4.4.113) may be increased by a factor of 1.2.

$$FS = 2.0 \qquad for \qquad F_{ic} \le 0.55S_y \tag{4.4.1}$$

$$FS = 2.407 - 0.741\left(\frac{F_{ic}}{S_y}\right) \qquad for \qquad 0.55S_y < F_{ic} < S_y \tag{4.4.2}$$

$$FS = 1.667 \qquad for \qquad F_{ic} = S_y \tag{4.4.3}$$

4.4.3 Material Properties

4.4.3.1 The equations for the allowable compressive stress are based on carbon and low alloy steel plate materials as given in Part 3. The maximum temperature limit permitted for these materials is defined in Table 4.4.1. For materials other than carbon or low alloy steel, a modification to the allowable stress is required. The procedure for modification of the allowable stress is to calculate the allowable compressive stress based on carbon and low alloy steel plate materials, and then make the following adjustments as described below.

a) Determine the tangent modulus, E_t, from paragraph 3.D.5 based on a stress equal to F_{xe}. For Axial Compression the allowable stress is adjusted as follows:

$$F_{xa} = \frac{F_{xe}}{FS}\frac{E_t}{E_y} \tag{4.4.4}$$

$$F_{ba} = F_{xa} \tag{4.4.5}$$

b) Determine the tangent modulus, E_t, from paragraph 3.D.5 based on a stress equal to F_{he}. For External Pressure the allowable stress is adjusted as follows:

$$F_{ha} = \frac{F_{he}}{FS}\frac{E_t}{E_y} \tag{4.4.6}$$

c) Determine the tangent modulus, E_t, from paragraph 3.D.5 based on a stress equal to F_{ve}. For Shear the allowable stress is adjusted as follows:

$$F_{va} = \frac{F_{ve}}{FS}\frac{E_t}{E_y} \tag{4.4.7}$$

4.4.3.2 The equations for the allowable compressive stress may be used in the time-independent region for the material of construction as provided in Table 4.4.1. If the component as designed is in the time-dependent region (i.e. creep is significant), the effects of time-dependent behavior shall be considered.

4.4.4 Shell Tolerances

4.4.4.1 Permissible Out-Of-Roundness Of Cylindrical, Conical, and Spherical Shells – The shell of a completed vessel subject to external pressure shall meet the following requirements at any cross section.

a) The out-of-roundness requirements in paragraph 4.3.2.1 shall be satisfied.

b) The maximum plus or minus deviation from a true circle, e, measured from a segmental circular template having the design inside or outside radius (depending on where the measurements are taken) and a chord length, L_{ec}, should not exceed the following value:

$$e = \min\left[e_c,\ 2t\right] \tag{4.4.8}$$

where

$$e_c = 0.0165t\left(\frac{L_{ec}}{\sqrt{Rt}}+3.25\right)^{1.069} \qquad valid \;\; for \;\; 0.2t \le e_c \le 0.0242R \tag{4.4.9}$$

$$L_{ec} = 2R\sin\left[\frac{\pi}{2n}\right] \tag{4.4.10}$$

$$n = \xi\left(\sqrt{\frac{R}{t}}\cdot\left(\frac{R}{L}\right)\right)^{\psi} \qquad valid \;\; for \;\; 2 \le n \le 1.41\sqrt{\frac{R}{t}} \tag{4.4.11}$$

$$\xi = \min\left[2.28\left(\frac{R}{t}\right)^{0.54} \;\; , \;\; 2.80\right] \tag{4.4.12}$$

$$\psi = \min\left[0.38\left(\frac{R}{t}\right)^{0.044} \;\; , \;\; 0.485\right] \tag{4.4.13}$$

c) The value of thickness, t, used in the above equations shall be determined as follows:

1) For vessels with butt joints, t is the nominal plate thickness less the corrosion allowance.
2) For vessels with lap joints, t is the nominal plate thickness and the permissible deviation is $e+t$.
3) Where the shell at any cross section is made from plates of different thicknesses t is the nominal plate thickness of the thinnest plate less the corrosion allowance.

d) For cones and conical sections, t shall be determined using paragraph 4.4.4.1.c except that t shall be replaced by t_c.

e) Measurements for out–of-tolerances shall be taken on the surface of the base metal and not on welds or other raised parts of the component.

f) The dimensions of a completed vessel may be brought within the requirements of this paragraph by any process that will not impair the strength of the material.

g) Sharp bends and flats spots shall not be permitted unless provision is made for them in the design or they satisfy the tolerances in paragraphs 4.4.4.2 and 4.4.4.4.

h) Vessels fabricated of pipe may have permissible variations in the outside diameter in accordance with those permitted under the specification covering its manufacture.

4.4.4.2 Cylindrical and Conical Shells Subject To Uniform Axial Compression and Axial Compression Due to a Bending Moment – the tolerance requirements in paragraph 4.3.2.1 shall be satisfied. In addition, the local inward deviation from a straight line, e, measured along a meridian over gauge length, L_x, shall not exceed the maximum permissible deviation, e_x, given below:

$$e_x = 0.002R_m \tag{4.4.14}$$

and,

$$L_x = \min\left[4\sqrt{R_m t},\ L\right] \qquad \textit{for cylindrical shells} \qquad (4.4.15)$$

$$L_x = \min\left[4\sqrt{\frac{R_m t_c}{\cos[\alpha]}},\ \frac{L_c}{\cos[\alpha]}\right] \qquad \textit{for conical shells} \qquad (4.4.16)$$

$$L_x = 25t \qquad \textit{across circumferential welds} \qquad (4.4.17)$$

4.4.4.3 Cylindrical and Conical Shells Subject To External Pressure And Uniform Axial Compression and Axial Compression Due to a Bending Moment – all of the tolerance requirements in paragraphs 4.4.4.1 and 4.4.4.2 shall be satisfied.

4.4.4.4 Spherical Shells and Formed Heads – the tolerance requirements in paragraph 4.3.2.2 shall be satisfied. In addition, the maximum local deviation from true circular form, e, for spherical shells and any spherical portion of a formed head shall not exceed the shell thickness. Measurements to determine the maximum local deviation shall be made with a template with a chord length, L_e, given by the following equation.

$$L_e = 3.72\sqrt{R_m t} \qquad (4.4.18)$$

4.4.4.5 Shells that do not meet the tolerance requirements of this paragraph may be evaluated using paragraph 4.14.

4.4.5 Cylindrical Shell

4.4.5.1 Required Thickness – The required thickness of a cylindrical shell subjected to external pressure loading shall be determined using the following procedure.

a) STEP 1 – Assume an initial thickness, t, and unsupported length, L (see Figures 4.4.1 and 4.4.2).

b) STEP 2 – Calculate the predicted elastic buckling stress, F_{he}.

$$F_{he} = \frac{1.6 C_h E_y t}{D_o} \qquad (4.4.19)$$

$$M_x = \frac{L}{\sqrt{R_o t}} \qquad (4.4.20)$$

$$C_h = 0.55\left(\frac{t}{D_o}\right) \qquad \textit{for } M_x \geq 2\left(\frac{D_o}{t}\right)^{0.94} \qquad (4.4.21)$$

$$C_h = 1.12 M_x^{-1.058} \qquad \textit{for } 13 < M_x < 2\left(\frac{D_o}{t}\right)^{0.94} \qquad (4.4.22)$$

$$C_h = \frac{0.92}{M_x - 0.579} \qquad \textit{for } 1.5 < M_x \leq 13 \qquad (4.4.23)$$

$$C_h = 1.0 \qquad for \;\; M_x \le 1.5 \tag{4.4.24}$$

c) STEP 3 – Calculate the predicted buckling stress, F_{ic}.

$$F_{ic} = S_y \qquad for \;\; \frac{F_{he}}{S_y} \ge 2.439 \tag{4.4.25}$$

$$F_{ic} = 0.7S_y\left(\frac{F_{he}}{S_y}\right)^{0.4} \qquad for \;\; 0.552 < \frac{F_{he}}{S_y} < 2.439 \tag{4.4.26}$$

$$F_{ic} = F_{he} \qquad for \;\; \frac{F_{he}}{S_y} \le 0.552 \tag{4.4.27}$$

d) STEP 4 – Calculate the value of design factor, FS, per paragraph 4.4.2.

e) STEP 5 – Calculate the allowable external pressure, P_a.

$$P_a = 2F_{ha}\left(\frac{t}{D_o}\right) \tag{4.4.28}$$

where,

$$F_{ha} = \frac{F_{ic}}{FS} \tag{4.4.29}$$

f) STEP 6 – If the allowable external pressure, P_a, is less than the design external pressure, increase the shell thickness or reduce the unsupported length of the shell (i.e. by the addition of a stiffening rings) and go to STEP 2. Repeat this process until the allowable external pressure is equal to or greater than the design external pressure.

4.4.5.2 Stiffening Ring Size – The following equations shall be used to determine the size of a stiffening ring.

a) Stiffening Ring Configuration – A combination of large and small stiffening rings may be used along the length of a shell. If a single size stiffener is used, then it shall be sized as a small stiffener. Alternatively, a combination of large and small stiffeners can be used to reduce the size of the intermittent small stiffening rings. The large stiffening rings may be sized to function as end stiffeners or bulkheads with small stiffeners spaced as required between end rings based on the shell thickness selected and loading combinations considered in the design.

b) Small Stiffening Ring – The required moment of inertia of the effective stiffening ring (i.e. actual stiffening ring plus the effective length of shell, see Figure 4.4.3) shall satisfy Equation (4.4.30). The parameter F_{he} shall be evaluated using the equations in paragraph 4.4.5.1 with $M_x = L_s \big/ \sqrt{R_o t}$.

$$I_s^C \ge \frac{1.5F_{he}L_sR_c^2t}{E_y\left(n^2-1\right)} \tag{4.4.30}$$

where,

$$n = \sqrt{\frac{2D_o^{1.5}}{3L_B t^{0.5}}} \quad where\ n\ is\ an\ integer;\ for\ n < 2\ use\ n = 2, \quad and\ \ for\ n > 10\ use\ n = 10 \tag{4.4.31}$$

The actual moment of inertia of the composite section comprised of the small stiffening ring and effective length of the shell about the centroidal axis shall be calculated using Equation (4.4.32):

$$I_s^C = I_s + A_s Z_s^2 \left(\frac{L_e t}{A_s + L_e t} \right) + \frac{L_e t^3}{12} \tag{4.4.32}$$

where,

$$L_e = 1.1\sqrt{D_o t} \tag{4.4.33}$$

c) Large Stiffening Ring or Bulkhead – The required moment of inertia of the effective stiffening ring (i.e. actual stiffening ring plus the effective length of shell) shall satisfy Equation (4.4.34). The parameter F_{hef} is the average value of the hoop buckling stress, F_{he}, over length L_F evaluated using the equations in paragraph 4.4.5.1 with $M_x = L_F \big/ \sqrt{R_o t}$.

$$I_L^C \geq \frac{F_{hef} L_F R_c^2 t}{2E_y} \tag{4.4.34}$$

The actual moment of inertia of the composite section comprised of the large stiffening ring and effective length of the shell about the centroidal axis shall be calculated using Equation (4.4.35):

$$I_L^C = I_L + A_L Z_L^2 \left(\frac{L_e t}{A_L + L_e t} \right) + \frac{L_e t^3}{12} \tag{4.4.35}$$

where,

$$L_e = 1.1\sqrt{D_o t}\left(\frac{A_s + L_s t}{A_L + L_s t} \right) \tag{4.4.36}$$

d) Local Stiffener Geometry Requirements for all Loading Conditions – The following equations shall be met to assure the stability of a stiffening ring.

1) Flat bar stiffener, flange of a tee section and the outstanding leg of an angle stiffener (see Figure 4.4.3)

$$\frac{h_1}{t_1} \leq 0.375\left(\frac{E_y}{S_y} \right)^{0.5} \tag{4.4.37}$$

2) Web of a tee stiffener or leg of an angle stiffener attached to the shell (see Figure 4.4.3).

$$\frac{h_2}{t_2} \leq \left(\frac{E_y}{S_y} \right)^{0.5} \tag{4.4.38}$$

e) Stiffener Size To Increase Allowable Longitudinal Compressive Stress – ring stiffeners can be used to increase the allowable longitudinal compressive stress for cylindrical or conical shells subject to uniform axial compression and axial compression due to bending. The required size of the stiffener shall satisfy the following equations. In addition, the spacing of the stiffeners must result in a value of $M_s \le 15$ where M_s is given by Equation (4.4.42).

$$A_s \ge \left(\frac{0.334}{M_s^{0.6}} - 0.063\right) L_s t \tag{4.4.39}$$

$$A_s \ge 0.06 L_s t \tag{4.4.40}$$

$$I_s^C \ge \frac{5.33 L_s t^3}{M_s^{1.8}} \tag{4.4.41}$$

$$M_s = \frac{L_s}{\sqrt{R_o t}} \tag{4.4.42}$$

f) Stiffener Size For Shear – The required size of the stiffener shall satisfy the following equation where C_v is evaluated using Equations (4.4.81) thru (4.4.84) with $M_x = M_s$, M_s is given by Equation (4.4.42).

$$I_s^C \ge 0.184 C_v M_s^{0.8} t^3 L_s \tag{4.4.43}$$

g) Arrangement Of Stiffening Rings

1) As shown in Figure 4.4.4, any joints between the ends or sections of such rings, at locations (A) and (B), and any connection between adjacent portions of a stiffening ring lying inside or outside the shell, at location (C), shall be made so that the required moment of inertia of the combined ring-shell section is maintained. For a section with a strut at location (D), the required moment of inertia shall be supplied by the strut alone.

2) As shown in Figure 4.4.4, stiffening rings placed on the inside of a vessel may be arranged as shown at locations (E) and (F) provided that the required moment of inertia of the ring at location (E) or of the combined ring-shell section at location (F) is maintained within the sections indicated. Where the gap at locations (A) or (E) does not exceed eight times the thickness of the shell plate, the combined moment of inertia of the shell and stiffener may be used.

3) Stiffening rings shall extend completely around the vessel except as provided below. Any gap in that portion of a stiffening ring supporting the shell, as shown in Figure 4.4.4 at locations (D) and (E), shall not exceed the length of arc given in Figure 4.4.5 unless additional reinforcement is provided as shown at location (C), or unless all of the following conditions are met:

   i) only one unsupported shell arc is permitted per ring

   ii) the length of unsupported shell arc does not exceed 90 deg

   iii) the unsupported shell arcs in adjacent stiffening rings are staggered 180 deg

   iv) the dimension L is taken as the larger of the distance between alternate stiffening rings or the distance from the head-bend line to the second stiffening ring plus one-third of the head depth

4) When internal plane structures perpendicular to the longitudinal axis of the cylinder, such as bubble trays or baffle plates, are used in a vessel, they may also be considered to act as stiffening rings provided they are designed to function as such.

5) Any internal stays or supports used shall bear against the shell of the vessel through the medium of a substantially continuous ring.

h) Attachment Of Stiffening Rings – Stiffening rings shall be attached to either the outside or the inside of the vessel by continuous welding, or if the component is not in cyclic service (i.e. a fatigue analysis is not required in accordance with paragraph 4.1.1.4) intermittent welding. Where gaps occur in the stiffening ring, the attachment weld shall conform to the details in paragraph 4.2.

4.4.5.3 Combined Loadings – cylindrical shells subject to external pressure and other loadings shall satisfy the requirements of paragraph 4.4.12.

4.4.6 Conical Shell

4.4.6.1 Required Thickness – The required thickness of a conical shell subjected to external pressure loading shall be determined using the equations for a cylinder by making the following substitutions:

a) The value of t_c is substituted for t in the equations in paragraph 4.4.5.

b) For offset cones, the cone angle, α, shall satisfy the requirements of paragraph 4.3.4.

c) The value of $0.5(D_L + D_S)/\cos[\alpha]$ is substituted for D_o in the equations in paragraph 4.4.5.

d) The value of $L_{ce}/\cos[\alpha]$ is substituted for L in the equations in paragraph 4.4.5 where L_{ce} is determined as shown below.

1) For Sketches (a) and (e) in Figure 4.4.7

$$L_{ce} = L_c \tag{4.4.44}$$

2) For Sketch (b) in Figure 4.4.7

$$L_{ce} = r_k \sin[\alpha] + L_c \tag{4.4.45}$$

3) For Sketch (c) in Figure 4.4.7

$$L_{ce} = r_f \sin[\alpha] + L_c \tag{4.4.46}$$

4) For Sketch (d) in Figure 4.4.7

$$L_{ce} = (r_k + r_f)\sin[\alpha] + L_c \tag{4.4.47}$$

e) Note that the half-apex angle of a conical transition can be computed knowing the shell geometry with the following equations. These equations were developed with the assumption that the conical transition contains a cone section, knuckle, or flare. If the transition does not contain a knuckle or flare, the radii of these components should be set to zero when computing the half-apex angle (see Figure 4.4.7).

1) If $(R_L - r_k) \geq (R_S + r_f)$:

$$\alpha = \beta + \phi \tag{4.4.48}$$

$$\beta = \arctan\left[\frac{(R_L - r_k) - (R_S + r_f)}{L_{ce}}\right] \tag{4.4.49}$$

2) If $(R_L - r_k) < (R_S + r_f)$:

$$\alpha = \phi - \beta \tag{4.4.50}$$

$$\beta = \arctan\left[\frac{\left(R_S + r_f\right) - \left(R_L - r_k\right)}{L_{ce}}\right] \tag{4.4.51}$$

3) In both cases shown above, the angle ϕ is given by the following equation.

$$\phi = \arcsin\left[\frac{(r_f + r_k)\cos\beta}{L_{ce}}\right] \tag{4.4.52}$$

4.4.6.2 Small Stiffening Rings – Intermediate circumferential stiffening rings within the conical transition shall be sized using Equation (4.4.30) where L_s is determined from paragraph 4.4.6.1.d, and t_c is the cone thickness at the ring location.

4.4.6.3 Combined Loadings – conical shells subject to external pressure and other loadings shall satisfy the requirements of paragraph 4.4.12.

4.4.7 Spherical Shell and Hemispherical Head

4.4.7.1 Required Thickness – The required thickness of a spherical shell or hemispherical head subjected to external pressure loading shall be determined using the following procedure.

a) STEP 1 – Assume an initial thickness, t for the spherical shell.

b) STEP 2 – Calculate the predicted elastic buckling stress, F_{he}.

$$F_{he} = 0.075E_y\left(\frac{t}{R_o}\right) \tag{4.4.53}$$

c) STEP 3 – Calculate the predicted buckling stress, F_{ic}.

$$F_{ic} = S_y \qquad for\ \frac{F_{he}}{S_y} \geq 6.25 \tag{4.4.54}$$

$$F_{ic} = \frac{1.31S_y}{\left(1.15 + \frac{S_y}{F_{he}}\right)} \qquad for\ 1.6 < \frac{F_{he}}{S_y} < 6.25 \tag{4.4.55}$$

$$F_{ic} = 0.18F_{he} + 0.45S_y \qquad for\ 0.55 < \frac{F_{he}}{S_y} \leq 1.6 \tag{4.4.56}$$

$$F_{ic} = F_{he} \qquad for \ \frac{F_{he}}{S_y} \le 0.55 \tag{4.4.57}$$

d) STEP 4 – Calculate the value of design margin, FS , per paragraph 4.4.2.

e) STEP 5 – Calculate the allowable external pressure, P_a

$$P_a = 2F_{ha}\left(\frac{t}{R_o}\right) \tag{4.4.58}$$

where,

$$F_{ha} = \frac{F_{ic}}{FS} \tag{4.4.59}$$

f) STEP 6 – If the allowable external pressure, P_a, is less than the design external pressure, increase the shell thickness and go to STEP 2. Repeat this process until the allowable external pressure is equal to or greater than the design external pressure.

4.4.7.2 Combined Loadings – spherical shells and hemispherical heads subject to external pressure and other loadings shall satisfy the requirements of paragraph 4.4.12.

4.4.8 Torispherical Head

4.4.8.1 Required Thickness – the required thickness of a torispherical head subjected to external pressure loading shall be determined using the equations for a spherical shell in paragraph 4.4.7 by substituting the outside crown radius for R_o .

4.4.8.2 Restrictions On Torispherical Head Geometry – the restriction of paragraph 4.3.6 shall apply.

4.4.8.3 Torispherical Heads With Different Dome and Knuckle Thicknesses – heads with this configuration shall be designed in accordance with Part 5.

4.4.8.4 Combined Loadings – torispherical heads subject to external pressure and other loadings shall satisfy the requirements of paragraph 4.4.12.

4.4.9 Ellipsoidal Head

4.4.9.1 Required Thickness – the required thickness of an elliptical head subjected to external pressure loading shall be determined using the equations for a spherical shell in paragraph 4.4.7 by substituting K_oD_o for R_o where K_o is given by the following equation:

$$K_o = 0.25346 + 0.13995\left(\frac{D_o}{2h_o}\right) + 0.12238\left(\frac{D_o}{2h_o}\right)^2 - 0.015297\left(\frac{D_o}{2h_o}\right)^3 \tag{4.4.60}$$

4.4.9.2 Combined Loadings – ellipsoidal heads subject to external pressure and other loadings shall satisfy the requirements of paragraph 4.4.12.

4.4.10 Local Thin Areas

Rules for the evaluation of Local Thin Areas are covered in paragraph 4.14.

4.4.11 Drilled Holes Not Penetrating Through the Vessel Wall

Design requirements for partially drilled holes that do not penetrate completely through the vessel wall are covered in paragraph 4.3.9.

4.4.12 Combined Loadings and Allowable Compressive Stresses

4.4.12.1 The rules in paragraphs 4.4.2 through 4.4.11 are applicable for external pressure loading. The rules in this paragraph provide allowable compressive stresses that shall be used for the design of shells subjected to supplemental loads that result in combined loadings. The allowable stresses of this paragraph shall also be used as the acceptance criteria for shells subjected to compressive stress evaluated using Part 5.

4.4.12.2 Cylindrical Shells – The allowable compressive stresses for cylindrical shells shall be computed using the following rules that are based on loading conditions. The loading conditions are underlined for clarity in the following paragraphs. Common parameters used in each of the loading conditions are given in paragraph 4.4.12.2.k.

a) <u>External Pressure Acting Alone</u> – the allowable hoop compressive membrane stress of a cylinder subject to external pressure acting alone, F_{ha}, is computed using the equations in paragraph 4.4.5.1.

b) <u>Axial Compressive Stress Acting Alone</u> – the allowable axial compressive membrane stress of a cylinder subject to an axial compressive load acting alone, F_{xa}, is computed using the following equations.

1) For $\lambda_c \le 0.15$ (Local Buckling):

$$F_{xa} = \min\left[F_{xa1},\ F_{xa2}\right] \tag{4.4.61}$$

$$F_{xa1} = \frac{S_y}{FS} \qquad for\ \frac{D_o}{t} \le 135 \tag{4.4.62}$$

$$F_{xa1} = \frac{466 S_y}{FS\left(331 + \dfrac{D_o}{t}\right)} \qquad for\ 135 < \frac{D_o}{t} < 600 \tag{4.4.63}$$

$$F_{xa1} = \frac{0.5 S_y}{FS} \qquad for\ \ 600 \le \frac{D_o}{t} \le 2000 \tag{4.4.64}$$

$$F_{xa2} = \frac{F_{xe}}{FS} \tag{4.4.65}$$

$$F_{xe} = \frac{C_x E_y t}{D_o} \tag{4.4.66}$$

$$C_x = \min\left[\frac{409\overline{c}}{\left(389+\frac{D_o}{t}\right)}, 0.9\right] \qquad for\ \frac{D_o}{t} < 1247 \tag{4.4.67}$$

$$C_x = 0.25\overline{c} \qquad for\ 1247 \le \frac{D_o}{t} \le 2000 \tag{4.4.68}$$

$$\overline{c} = 2.64 \qquad for\ M_x \le 1.5 \tag{4.4.69}$$

$$\overline{c} = \frac{3.13}{M_x^{0.42}} \qquad for\ 1.5 < M_x < 15 \tag{4.4.70}$$

$$\overline{c} = 1.0 \quad for \quad M_x \ge 15 \tag{4.4.71}$$

2) For $\lambda_c > 0.15\ and\ K_u L_u / r_g < 200$ (Column Buckling):

$$F_{ca} = F_{xa}\left[1-0.74\left(\lambda_c - 0.15\right)\right]^{0.3} \qquad for\ 0.15 < \lambda_c < 1.147 \tag{4.4.72}$$

$$F_{ca} = \frac{0.88F_{xa}}{\lambda_c^2} \qquad for\ \lambda_c \ge 1.147 \tag{4.4.73}$$

c) Compressive Bending Stress – the allowable axial compressive membrane stress of a cylindrical shell subject to a bending moment acting across the full circular cross section F_{ba}, is computed using the following equations.

$$F_{ba} = F_{xa} \qquad for\ 135 \le \frac{D_o}{t} \le 2000 \tag{4.4.74}$$

$$F_{ba} = \frac{466S_y}{FS\left(331+\frac{D_o}{t}\right)} \qquad for\ 100 \le \frac{D_o}{t} < 135 \tag{4.4.75}$$

$$F_{ba} = \frac{1.081S_y}{FS} \qquad for\ \frac{D_o}{t} < 100\ and\ \gamma \ge 0.11 \tag{4.4.76}$$

$$F_{ba} = \frac{S_y\left(1.4-2.9\gamma\right)}{FS} \qquad for\ \frac{D_o}{t} < 100\ and\ \gamma < 0.11 \tag{4.4.77}$$

$$\gamma = \frac{S_y D_o}{E_y t} \tag{4.4.78}$$

d) Shear Stress – the allowable shear stress of a cylindrical shell , F_{va}, is computed using the following equations.

$$F_{va} = \frac{\eta_v F_{ve}}{FS} \tag{4.4.79}$$

$$F_{ve} = \alpha_v C_v E_y \left(\frac{t}{D_o}\right) \tag{4.4.80}$$

$$C_v = 4.454 \qquad for\ M_x \le 1.5 \tag{4.4.81}$$

$$C_v = \left(\frac{9.64}{M_x^2}\right)\left(1+0.0239M_x^3\right)^{0.5} \qquad for\ 1.5 < M_x < 26 \tag{4.4.82}$$

$$C_v = \frac{1.492}{M_x^{0.5}} \qquad for\ 26 \le M_x < 4.347\left(\frac{D_o}{t}\right) \tag{4.4.83}$$

$$C_v = 0.716\left(\frac{t}{D_o}\right)^{0.5} \qquad for\ M_x \ge 4.347\left(\frac{D_o}{t}\right) \tag{4.4.84}$$

$$\alpha_v = 0.8 \qquad for\ \frac{D_o}{t} \le 500 \tag{4.4.85}$$

$$\alpha_v = 1.389 - 0.218\log_{10}\left(\frac{D_o}{t}\right) \qquad for\ \frac{D_o}{t} > 500 \tag{4.4.86}$$

$$\eta_v = 1.0 \qquad for\ \frac{F_{ve}}{S_y} \le 0.48 \tag{4.4.87}$$

$$\eta_v = 0.43\left(\frac{S_y}{F_{ve}}\right) + 0.1 \qquad for\ 0.48 < \frac{F_{ve}}{S_y} < 1.7 \tag{4.4.88}$$

$$\eta_v = 0.6\left(\frac{S_y}{F_{ve}}\right) \qquad for\ \frac{F_{ve}}{S_y} \ge 1.7 \tag{4.4.89}$$

e) Axial Compressive Stress And Hoop Compression – the allowable compressive stress for the combination of uniform axial compression and hoop compression, F_{xha}, is computed using the following equations:

1) For $\lambda_c \le 0.15$; F_{xha} is computed using the following equation with F_{ha} and F_{xa} evaluated using the equations in paragraphs 4.4.12.2.a and 4.4.12.2.b.1, respectively.

$$F_{xha} = \left[\left(\frac{1}{F_{xa}^2} \right) - \left(\frac{C_1}{C_2 F_{xa} F_{ha}} \right) + \left(\frac{1}{C_2^2 F_{ha}^2} \right) \right]^{-0.5} \tag{4.4.90}$$

$$C_1 = \frac{\left(F_{xa} \cdot FS + F_{ha} \cdot FS \right)}{S_y} - 1.0 \tag{4.4.91}$$

$$C_2 = \frac{f_x}{f_h} \tag{4.4.92}$$

$$f_x = f_a + f_q \qquad\qquad for\ f_x \le F_{xha} \tag{4.4.93}$$

The parameters f_a and f_q are defined in paragraph 4.4.12.2.k.

2) For $0.15 < \lambda_c \le 1.2$: F_{xha} is computed from the following equation with $F_{ah1} = F_{xha}$ evaluated using the equations in paragraph 4.4.12.2.e.1 with $f_x = f_a$, and F_{ca} evaluated using the equations in paragraph 4.4.12.2.b.2. As noted, the load on the end of a cylinder due to external pressure does not contribute to column buckling and therefore F_{ah1} is compared with f_a rather than f_x. The stress due to the pressure load does, however, lower the effective yield stress and the quantity in $\left(1 - f_q / S_y\right)$ accounts for this reduction

$$F_{xha} = \min\left[F_{ah1},\, F_{ah2}\right] \tag{4.4.94}$$

$$F_{ah2} = F_{ca} \left(1 - \frac{f_q}{S_y} \right) \tag{4.4.95}$$

3) For $\lambda_c \le 0.15$, the allowable hoop compressive membrane stress, F_{hxa}, is given by the following equation:

$$F_{hxa} = \frac{F_{xha}}{C_2} \tag{4.4.96}$$

f) Compressive Bending Stress And Hoop Compression – the allowable compressive stress for the combination of axial compression due to a bending moment and hoop compression, F_{bha}, is computed using the following equations.

1) An iterative solution procedure is utilized to solve these equations for C_3 with F_{ha} and F_{ba} evaluated using the equations in paragraphs 4.4.12.2.a and 4.4.12.2.c, respectively.

$$F_{bha} = C_3 C_4 F_{ba} \tag{4.4.97}$$

$$C_4 = \left(\frac{f_b}{f_h} \right) \left(\frac{F_{ha}}{F_{ba}} \right) \tag{4.4.98}$$

$$C_3^2\left(C_4^2+0.6C_4\right)+C_3^{2n}-1=0 \tag{4.4.99}$$

$$n=5-\frac{4F_{ha}\cdot FS}{S_y} \tag{4.4.100}$$

2) The allowable hoop compressive membrane stress, F_{hba}, is given by the following equation:

$$F_{hba}=F_{bha}\left(\frac{f_h}{f_b}\right) \tag{4.4.101}$$

g) Shear Stress And Hoop Compression – The allowable compressive stress for the combination of shear, F_{vha}, and hoop compression is computed using the following equations.

1) The allowable shear stress is given by the following equation with F_{ha} and F_{va} evaluated using the equations in paragraphs 4.4.12.2.a and 4.4.12.2.d, respectively.

$$F_{vha}=\left[\left(\frac{F_{va}^2}{2C_5F_{ha}}\right)^2+F_{va}^2\right]^{0.5}-\frac{F_{va}^2}{2C_5F_{ha}} \tag{4.4.102}$$

$$C_5=\frac{f_v}{f_h} \tag{4.4.103}$$

2) The allowable hoop compressive membrane stress, F_{hva}, is given by the following equation:

$$F_{hva}=\frac{F_{vha}}{C_5} \tag{4.4.104}$$

h) Axial Compressive Stress, Compressive Bending Stress, Shear Stress, And Hoop Compression – The allowable compressive stress for the combination of uniform axial compression, axial compression due to a bending moment, and shear in the presence of hoop compression is computed using the following interaction equations.

1) The shear coefficient is determined using the following equation with F_{va} from paragraph 4.4.12.2.d.

$$K_s=1.0-\left(\frac{f_v}{F_{va}}\right)^2 \tag{4.4.105}$$

2) For $\lambda_c \le 0.15$; the acceptability of a member subject to compressive axial and bending stresses, f_a and f_b, respectively, is determined using the following interaction equation with F_{xha} and F_{bha} evaluated using the equations in paragraphs 4.4.12.2.e.1 and 4.4.12.f.1, respectively.

$$\left(\frac{f_a}{K_sF_{xha}}\right)^{1.7}+\left(\frac{f_b}{K_sF_{bha}}\right)\le 1.0 \tag{4.4.106}$$

3) For $0.15 < \lambda_c \le 1.2$; the acceptability of a member subject to compressive axial and bending stresses, f_a and f_b, respectively, is determined using the following interaction equation with F_{xha} and F_{bha} evaluated using the equations in paragraphs 4.4.12.2.e.2 and 4.4.12.2.f.1, respectively.

$$\left(\frac{f_a}{K_s F_{xha}}\right)+\left(\frac{8}{9}\frac{\Delta f_b}{K_s F_{bha}}\right)\le 1.0 \qquad for \quad \frac{f_a}{K_s F_{xha}} \ge 0.2 \tag{4.4.107}$$

$$\left(\frac{f_a}{2K_s F_{xha}}\right)+\left(\frac{\Delta f_b}{K_s F_{bha}}\right)\le 1.0 \qquad for \quad \frac{f_a}{K_s F_{xha}} < 0.2 \tag{4.4.108}$$

$$\Delta = \frac{C_m}{1-\left(\dfrac{f_a \cdot FS}{F_e}\right)} \tag{4.4.109}$$

$$F_e = \frac{\pi^2 E_y}{\left(\dfrac{K_u L_u}{r_g}\right)^2} \tag{4.4.110}$$

i) <u>Axial Compressive Stress, Compressive Bending Stress, And Shear</u> – The allowable compressive stress for the combination of uniform axial compression, axial compression due to a bending moment, and shear in the absence of hoop compression is computed using the following interaction equations:

1) The shear coefficient is determined using the equation in paragraph 4.4.12.2.h.1 with F_{va} from paragraph 4.4.12.2.d.

2) For $\lambda_c \le 0.15$; the acceptability of a member subject to compressive axial and bending stresses f_a and f_b, respectively, is determined using the following interaction equation with, F_{xa} and F_{ba} evaluated using the equations in paragraphs 4.4.12.2.b.1 and 4.4.12.2.c, respectively

$$\left(\frac{f_a}{K_s F_{xa}}\right)^{1.7}+\left(\frac{f_b}{K_s F_{ba}}\right)\le 1.0 \tag{4.4.111}$$

3) For $0.15 < \lambda_c \le 1.2$; the acceptability of a member subject to compressive axial and bending stresses, f_a and f_b, respectively, is determined using the following interaction equation with, F_{ca} and F_{ba} evaluated using the equations in paragraphs 4.4.12.2.b.2 and 4.4.12.2.c respectively. The coefficient Δ is evaluated using the equations in paragraph 4.4.12.2.h.3

$$\left(\frac{f_a}{K_s F_{ca}}\right)+\left(\frac{8}{9}\frac{\Delta f_b}{K_s F_{ba}}\right)\le 1.0 \qquad for \quad \frac{f_a}{K_s F_{ca}} \ge 0.2 \tag{4.4.112}$$

$$\left(\frac{f_a}{2K_sF_{ca}}\right)+\left(\frac{\Delta f_b}{K_sF_{ba}}\right)\le 1.0 \qquad for \;\; \frac{f_a}{K_sF_{ca}}<0.2 \tag{4.4.113}$$

j) The maximum deviation, e may exceed the value of e_x given in paragraph 4.4.4.2 if the maximum axial stress is less than F_{xa} for shells designed for axial compression only, or less than F_{xha} for shells designed for combinations of axial compression and external pressure. The change in buckling stress, F'_{xe}, is given by Equation (4.4.114). The reduced allowable buckling stress, $F_{xa(reduced)}$, is determined using Equation (4.4.115) where e is the new maximum deviation, F_{xa} is determined using Equation (4.4.61), and FS_{xa} is the value of the stress reduction factor used to determine F_{xa}.

$$F'_{xe}=\left(0.944-\left|0.286\log\left[\frac{0.0005e}{e_x}\right]\right|\right)\left(\frac{E_yt}{R}\right) \tag{4.4.114}$$

$$F_{xa(reduced)}=\frac{F_{xa}\cdot FS_{xa}-F'_{xe}}{FS_{xa}} \tag{4.4.115}$$

The quantity $0.286\log\left[0.0005(e/e_x)\right]$ in Equation (4.4.114) is an absolute number (i.e. the log of a very small number is negative). For example, if $e=e_x$, then the change in the buckling stress computed using Equation (4.4.114) is $F'_{xe}=0.086E_y(t/R)$.

k) Section Properties, Stresses, Buckling Parameters – equations for section properties, nominal shell stresses, and buckling parameters that are used in paragraphs 4.12.2.a through 4.12.2.i are provided below.

$$A=\frac{\pi\left(D_o^2-D_i^2\right)}{4} \tag{4.4.116}$$

$$S=\frac{\pi\left(D_o^4-D_i^4\right)}{32D_o} \tag{4.4.117}$$

$$f_h=\frac{PD_o}{2t} \tag{4.4.118}$$

$$f_b=\frac{M}{S} \tag{4.4.119}$$

$$f_a=\frac{F}{A} \tag{4.4.120}$$

$$f_q=\frac{P\pi D_i^2}{4A} \tag{4.4.121}$$

$$f_v = \frac{V\sin[\phi]}{A} \tag{4.4.122}$$

$$r_g = 0.25\sqrt{D_o^2 + D_i^2} \tag{4.4.123}$$

$$M_x = \frac{L}{\sqrt{R_o t}} \tag{4.4.124}$$

$$\lambda_c = \frac{K_u L_u}{\pi\, r_g}\left(\frac{F_{xa}\cdot FS}{E_y}\right)^{0.5} \tag{4.4.125}$$

4.4.12.3 Conical Shells – Unstiffened conical transitions or cone sections between stiffening rings of conical shells with a half-apex angle, α, less than 60° shall be evaluated as an equivalent cylinder using the equations in paragraph 4.4.12.2 with the substitutions shown below. Both the shell tolerances and stress criteria in this paragraph shall be satisfied at all cross-sections along the length of the cone.

a) The value of t_c is substituted for t to determine the allowable compressive stress.

b) The value of $D/\cos\alpha$ is substituted for D_o to determine the allowable compressive stress where D is the outside diameter of the cone at the point under consideration.

c) The value of $L_c/\cos\alpha$, is substituted for L where L_c is the distance along the cone axis between stiffening rings.

4.4.12.4 Spherical Shells and Formed Heads – The allowable compressive stresses are based on the ratio of the biaxial stress state.

a) Equal Biaxial Stresses – The allowable compressive stress for a spherical shell subject to a uniform external pressure, F_{ha}, is given by the equations in paragraph 4.4.7.

b) Unequal Biaxial Stresses, Both Stresses Are Compressive – The allowable compressive stress for a spherical shell subject to unequal biaxial stresses, σ_1 and σ_2, where both σ_1 and σ_2 are compressive stresses resulting from the applied loads is given by the equations shown below. In these equations, F_{ha} is determined using paragraph 4.4.7. F_{1a} is the allowable compressive stress in the direction of σ_1 and F_{2a} is the allowable compressive stress in the direction of σ_2.

$$F_{1a} = \frac{0.6F_{ha}}{1-0.4k} \tag{4.4.126}$$

$$F_{2a} = kF_{1a} \tag{4.4.127}$$

$$k = \frac{\sigma_2}{\sigma_1} \qquad where\ |\sigma_1| > |\sigma_2| \tag{4.4.128}$$

c) Unequal Biaxial Stresses, One Stress Is Compressive And The Other Is Tensile – The allowable compressive stress for a spherical shell subject to unequal biaxial stresses, σ_1 and σ_2, where σ_1 is

compressive and σ_2 is tensile resulting from the applied loads is given by the equations shown below. In these equations, F_{1a} is the allowable compressive stress in the direction of σ_1, and is the value of F_{ha} determined using paragraph 4.4.7 with F_{he} computed using the following equations.

$$F_{he} = \frac{\left(C_o + C_p\right) E_y t}{R_o} \tag{4.4.129}$$

$$C_o = \frac{102.2}{195 + \frac{R_o}{t}} \qquad for \quad \frac{R_o}{t} < 622 \tag{4.4.130}$$

$$C_o = 0.125 \qquad for \quad 622 \le \frac{R_o}{t} \le 1000 \tag{4.4.131}$$

$$C_p = \frac{1.06}{3.24 + \left(\frac{E_y t}{\sigma_2 R_o}\right)} \tag{4.4.132}$$

4.4.13 Cylindrical-To-Conical Shell Transition Junctions Without A Knuckle

4.4.13.1 The design rules in paragraph 4.3.11 shall be satisfied. In these calculations, a negative value of pressure shall be used in all applicable equations.

4.4.13.2 If a stiffening ring is provided at the cone-to-cylinder junction, the design shall be made in accordance with Part 5.

4.4.14 Cylindrical-To-Conical Shell Transition Junctions With A Knuckle

4.4.14.1 The design rules in paragraph 4.3.12 shall be satisfied. In these calculations, a negative value of pressure shall be used in all applicable equations.

4.4.14.2 If a stiffening ring is provided within the knuckle, the design shall be made in accordance with Part 5.

4.4.15 Nomenclature

A cross-sectional area of cylinder.

A_s cross-sectional area of a small ring stiffener.

A_L cross-sectional area of a large ring stiffener that acts as a bulkhead.

α one-half of the conical shell apex angle (degrees).

C_m coefficient whose value is established as follows:

= 0.85 for compression members in frames subject to joint translation (sideway).

= $0.6 - 0.4\left(M_1/M_2\right)$ for rotationally restrained members in frames braced against joint translation and not subject to transverse loading between their supports in the plane of bending; in this equation, M_1/M_2 is the ratio of the smaller to large bending moment at the ends of the portion of the member that is unbraced in the plane of bending under

consideration (M_1/M_2 is positive when the member is bent in reverse curvature and negative when the member is bent in single curvature).
= 0.85 for compression members in frames braced against joint translation and subject to transverse loading between support points, the member ends are restrained against rotation in the plane of bending.
= 1.0 for compression members in frames braced against joint translation and subject to transverse loading between support points, the member ends are unrestrained against rotation in the plane of bending.
= 1.0 for an unbraced skirt supported vessel.

c	distance from the neutral axis to the point under consideration.
D_c	diameter to the centroid of the composite ring section for an external ring; or the inside diameter for and internal ring (see Figure 4.4.6)
D_i	Inside diameter of cylinder (including the effects of corrosion.
D_o	outside diameter of cylinder.
D_e	outside diameter of an assumed equivalent cylinder for the design of cones or conical sections.
D_S	outside diameter of at the small end of the cone or conical section between lines of support.
D_L	outside diameter of at the large end of the cone or conical section between lines of support.
E_y	modulus of elasticity of material at the design temperature from Part 3 .
E_t	tangent modulus of elasticity of material at the design temperature from Part 3 .
F	applied net-section axial compression load.
f_a	axial compressive membrane stress resulting from applied axial load.
f_b	axial compressive membrane stress resulting from applied bending moment.
f_h	hoop compressive stress in the cylinder from external pressure.
f_q	axial compressive membrane stress resulting from the pressure load, Q_p, on the end of the cylinder.
f_v	shear stress from applied loads.
FS	design factor.
F_{ba}	allowable compressive membrane stress of a cylinder subject to a net-section bending moment in the absence of other loads.
F_{ca}	allowable compressive membrane stress of a cylinder due to an axial compressive load with $\lambda_c = 0.15$.
F_{bha}	allowable axial compressive membrane stress of a cylinder subject to bending in the presence of hoop compression.
F_{hba}	allowable hoop compressive membrane stress of a cylinder in the presence of longitudinal compression due to net-section bending moment.
F_{he}	elastic hoop compressive membrane failure stress of a cylinder or formed head subject to external pressure only.
F_{ha}	allowable hoop compressive membrane stress of a cylinder or formed head subject to external pressure only.
F_{hef}	average value of the hoop buckling stress, F_{he}, averaged over the length L_F where F_{he} is determined from Equation (4.4.19).

F_{hva} allowable hoop compressive membrane stress of a cylinder in the presence of shear stress.

F_{hxa} allowable hoop compressive membrane stress of a cylinder in the presence of axial compression.

F_{ic} predicted buckling stress, which is determined by letting $FS = 1.0$ in the allowable stress equations.

F_{ta} allowable tensile stress from paragraph 3.8.

F_{va} allowable shear stress of a cylinder subject only to shear loads.

F_{ve} elastic shear buckling stress of a cylinder subject only to shear loads.

F_{vha} allowable shear stress of a cylinder subject to shear stress in the presence of hoop compression.

F_{xa} allowable compressive membrane stress of a cylinder due to an axial compressive load with $\lambda_c \le 0.15$.

F_{xe} elastic axial compressive failure membrane stress (local buckling) of a cylinder in the absence of other loads.

F_{xha} allowable axial compressive membrane stress of a cylinder in the presence of hoop compression for $\lambda_c \le 0.15$.

γ Buckling parameter.

h_o height of the elliptical head measured to the outside surface.

h_1 length of a flat bar stiffener, or leg of an angle stiffener, or flange of a tee stiffener, as applicable.

h_2 length of the angle leg or web of the stiffener, as applicable.

I moment of inertia of the cylinder or cone cross section.

I_L actual moment of inertia of the large stiffening ring.

I_L^C actual moment of inertia of the composite section comprised of the large stiffening ring and effective length of the shell about the centroidal axis.

I_s actual moment of inertia of the small stiffening ring.

I_s^C actual moment of inertia of the composite section comprised of the small stiffening ring and effective length of the shell about the centroidal axis.

K_o elliptical head factor:

K_u coefficient based on end conditions of a member subject to axial compression:

= 2.10 for a member with one free end and the other end fixed. In this case, “member” is the unbraced cylindrical shell or cylindrical shell section as defined in the Nomenclature.
= 1.00 for a member with both ends pinned,
= 0.80 for a member with one end pinned and the other end fixed,
= 0.65 for a member with both ends fixed.

$L, L_1, L_2, ...$ design lengths of the unstiffened vessel sections between lines of support (see Figure 4.4.2). A line of support is (1) a circumferential line on a head (excluding conical heads) at one-third the depth of the head measured from the tangent line, (2) a small stiffening ring that meets the requirements of paragraph 4.4.5.2.b, or (3) a tubesheet.

$L_B, L_{B1}, L_B, ...$ design lengths of the cylinder between bulkheads or large rings designated to act as bulkheads (see Figure 4.4.2).

L_c axial length of a cone or conical section for an unstiffened cone, or the length from the cone-to-cylinder junction to the first stiffener in the cone for a stiffened cone (see Figures 4.4.6 and 4.4.7).

L_e effective length of the shell.

L_F one-half of the sum of the distances, L_B, from the center line of a large ring to the next large ring of head line of support on either side of the large ring.

L_s one-half of the sum of the distances from the centerline of a stiffening ring to the next line of support on either side of the ring measured parallel to the axis of the cylinder. A line of support is defined in the definition of L.

L_t overall length of the vessel.

L_u laterally unbraced length of cylindrical member that is subject to column buckling, equal to zero when evaluating the shell of a vessel under external pressure only.

λ_c slenderness factor for column buckling.

M applied net-section bending moment.

M_x shell parameter.

P applied external pressure.

P_a allowable external pressure in the absence of other loads.

ϕ angle measured around the circumference from the direction of the applied shear force to the point under consideration.

r_f inside radius of the flare.

r_g radius of gyration.

r_k inside radius of the knuckle.

R inside or outside radius of cylindrical, conical, and spherical shells, as applicable

R_m radius to the centerline of the shell.

R_c radius to the centroid of the combined ring stiffener and effective length of the shell, $R_c = R + Z_c$ (see Figure 4.4.3)

R_L inside radius of the cylinder at the large end of a cone to cylinder junction.

R_o outside radius of a cylinder or spherical shell.

R_S inside radius of the cylinder at the small end of a cone to cylinder junction.

S section modulus of the shell.

S_y minimum specified yield strength from Annex 3.D at specified design metal temperature.

σ_1 principal compressive stress in the 1-direction.

σ_2 principal compressive stress in the 2-direction.

V net-section shear force.

t shell thickness.

t_c cone thickness.

t_L shell thickness of large end cylinder at a conical transition.

t_S shell thickness of small end cylinder at a conical transition.

t_1 thickness of a flat bar stiffener, or leg of an angle stiffener, or flange of a tee stiffener, as applicable.

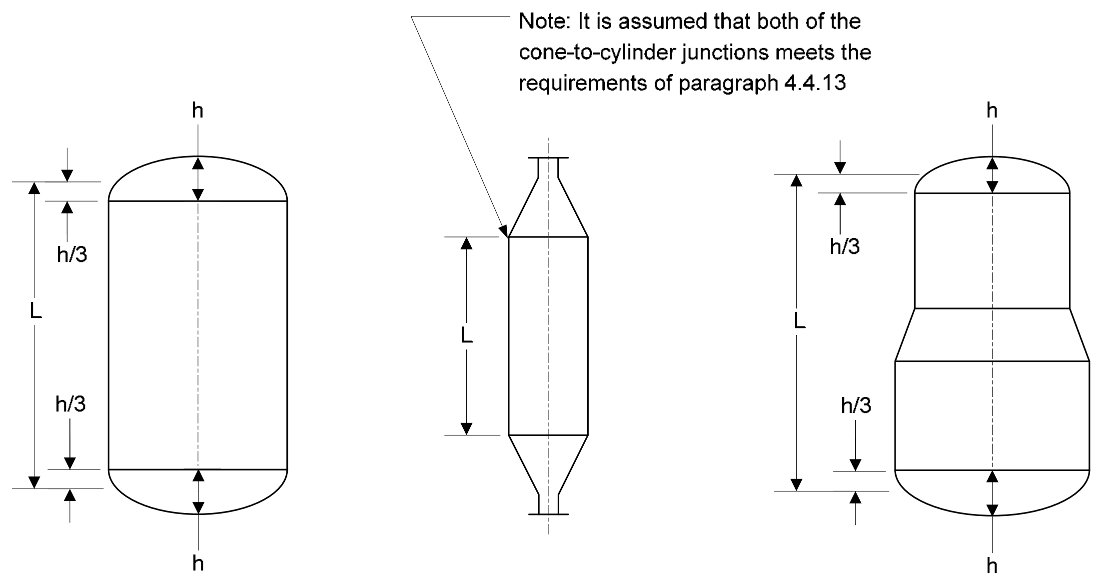
t_2	thickness of the angle leg or web of the stiffener, as applicable.
Z_c	radial distance from the centerline of the shell to the combined section of the ring stiffener and effective length of the shell.
Z_L	radial distance from the centerline of the shell to the centroid of the large ring stiffener.
Z_s	radial distance from the centerline of the shell to the centroid of the small ring stiffener.

4.4.16 Tables

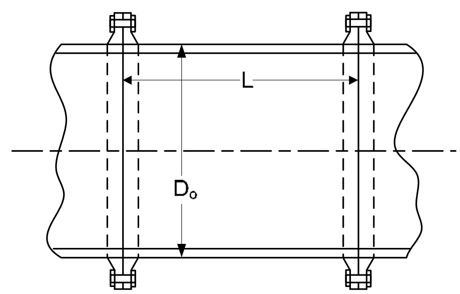
Table 4.4.1 – Maximum Metal Temperature For Compressive Stress Rules

Materials	Temperature Limit	
	°C	°F
Carbon and Low Alloy Steels - Table 3.A.1	425	800
High Alloy Steels - Table 3.A.3	425	800
Quenched and Tempered Steels - Table 3.A.2	370	700
Aluminum and Aluminum Alloys - Table 3.A.4	150	300
Copper and Copper Alloys - Table 3.A.5	65	150
Nickel and Nickel Alloys - Table 3.A.6	480	900
Titanium and Titanium Alloys - Table 3.A.7	315	600

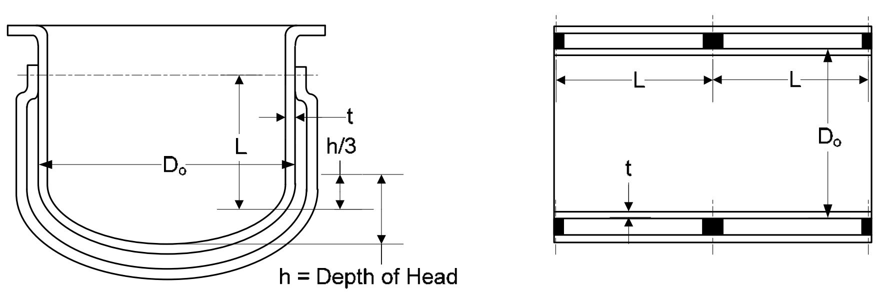
4.4.17 Figures



(a) Typical Unsupported Length Dimensions On Vessel Without Stiffeners



(b) Typical Unsupported Length Between Flange Pairs



(c) Typical Unsupported Length For Jacketed Vessels

Figure 4.4.1 – Lines of Support or Unsupported Length for Typical Vessel Configurations

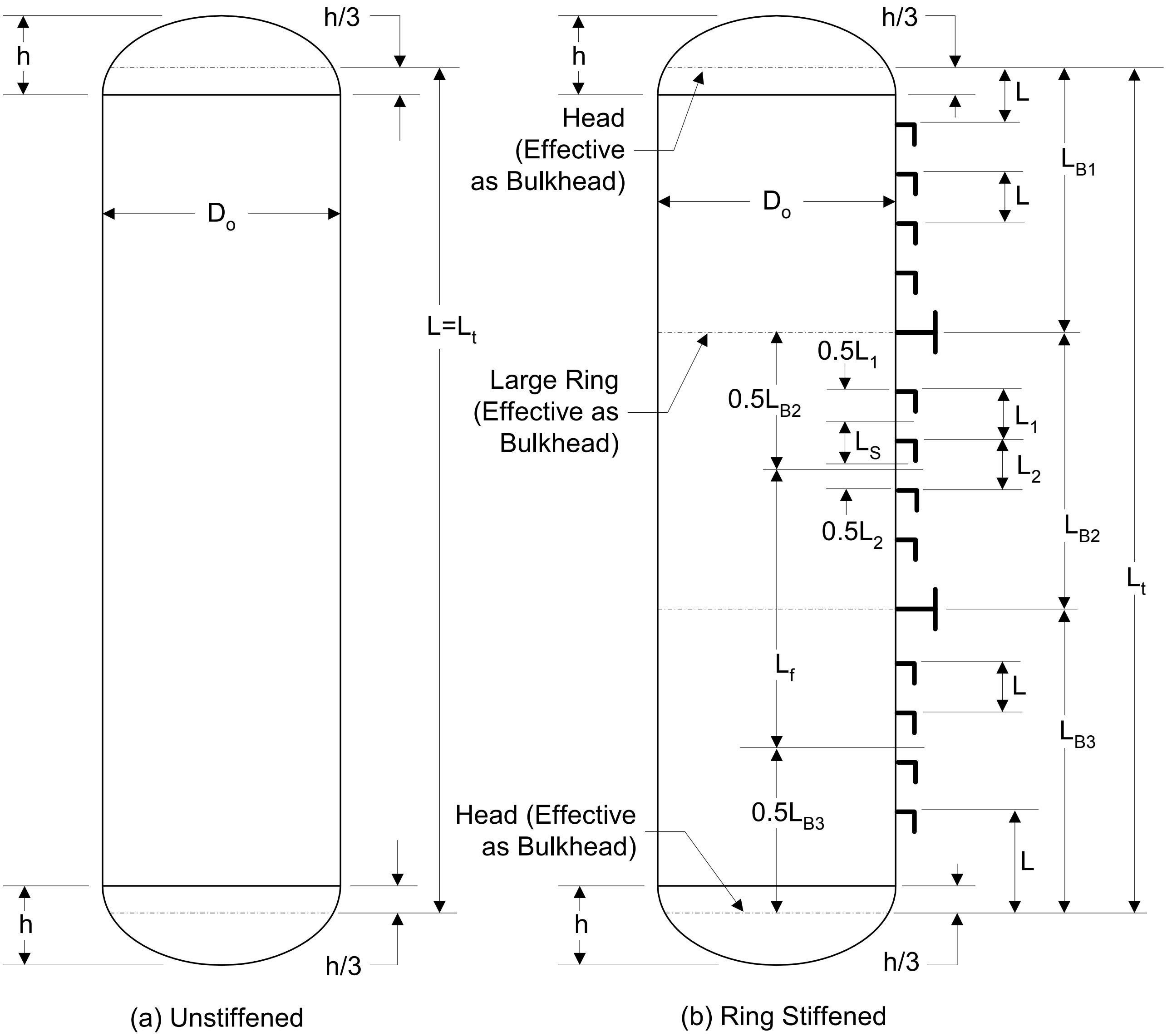
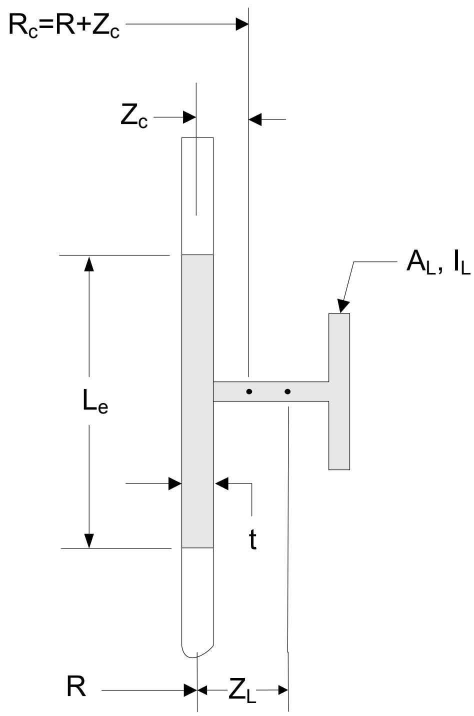
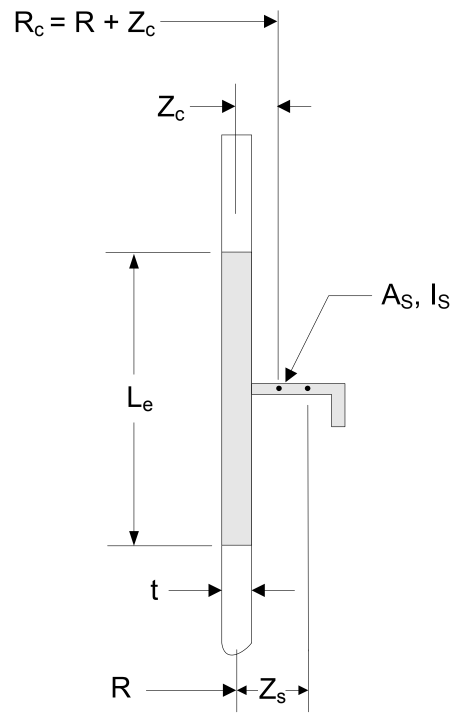


Figure 4.4.2 – Lines of Support or Unsupported Length for Unstiffened and Stiffened Cylindrical Shells

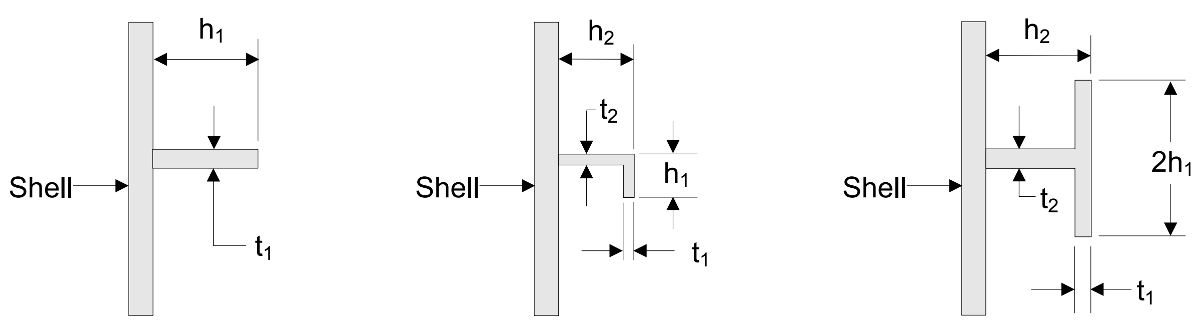


(a-1) Stiffening Ring which Acts as a Bulkhead



(a-2) Small Stiffening Ring

(a) Sections Through Stiffening Rings



(b) Stiffener Variables for Local Buckling Calculation

Figure 4.4.3 – Stiffener Ring Parameters

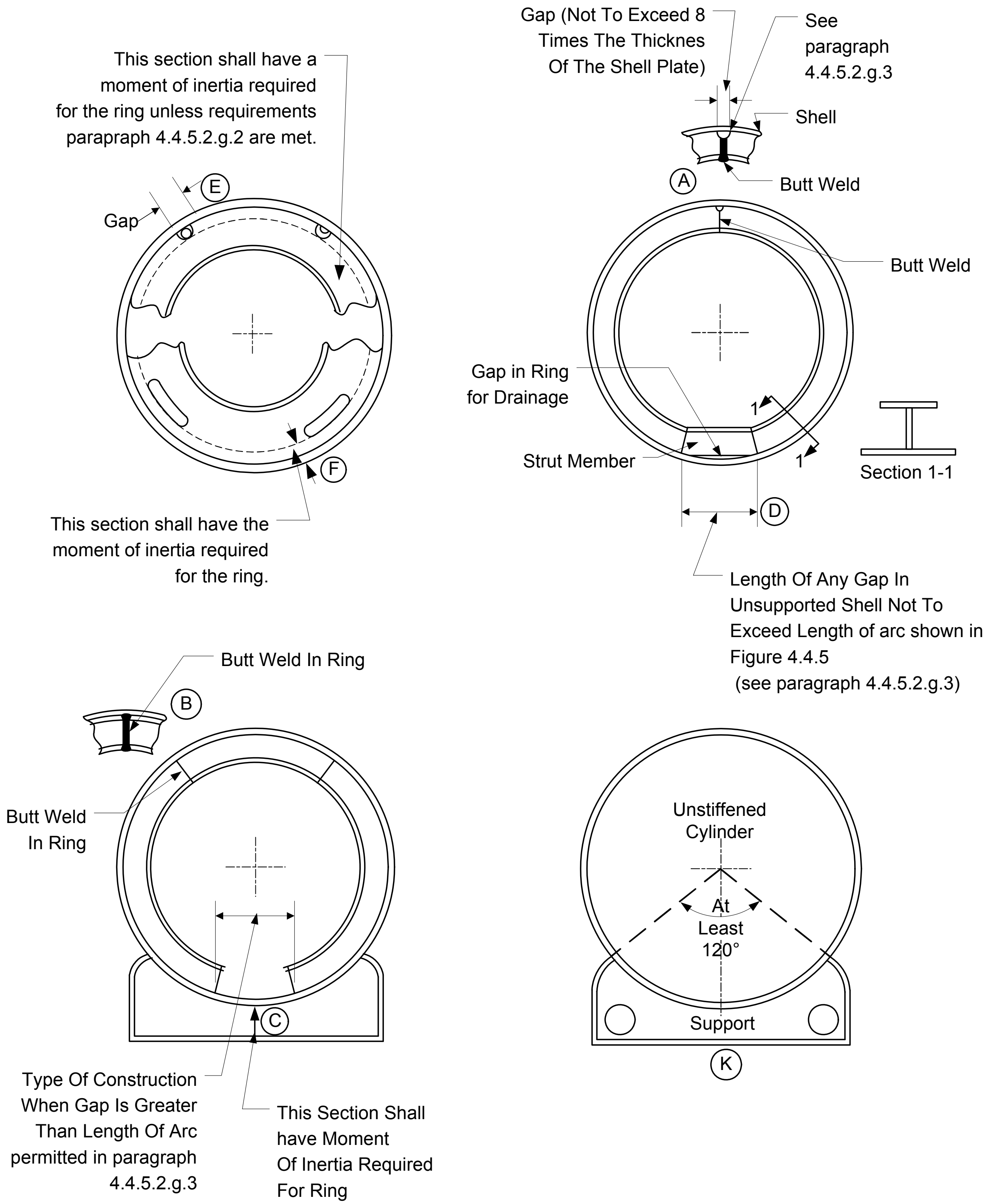
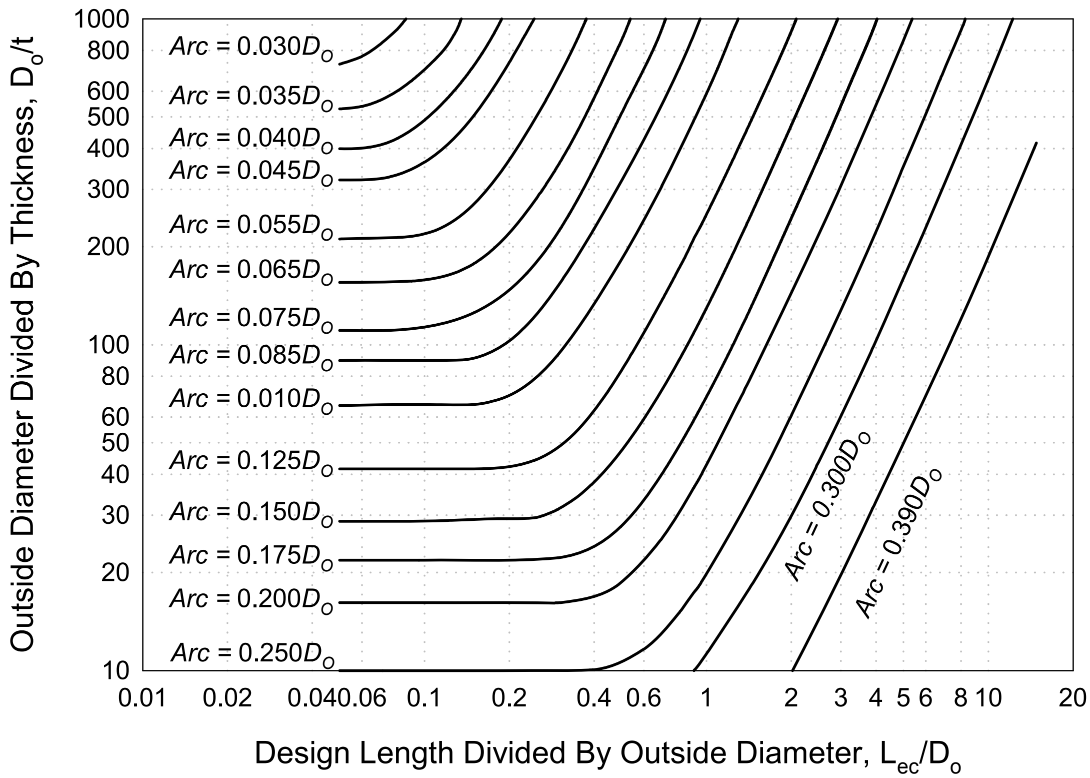


Figure 4.4.4 – Various Arrangements of Stiffening Rings for Cylindrical Vessels Subjected to External Pressure



Notes:

1. Cylindrical Shells – L_{ec} is the unsupported length of the cylinder and D_o is the outside diameter.
2. Conical Shells – L_{ec} and D_o are established using the following equations for any cross section having a diameter D_x. In these equations D_L and D_S are the cone large end and small end outside diameters, respectively and L is the unsupported length of the conical section under evaluation.

$$L_{ec} = \left(\frac{L}{2}\right)\left(1+\frac{D_S}{D_L}\right)\left(\frac{D_S}{D_L}\right) \tag{4.4.133}$$

$$D_o = \frac{0.5\left(D_L + D_S\right)}{\cos\left[\alpha\right]} \tag{4.4.134}$$

Figure 4.4.5 – Maximum Arc of Shell Left Unsupported Because of a Gap in the Stiffening Ring of a Cylindrical Shell Under External Pressure

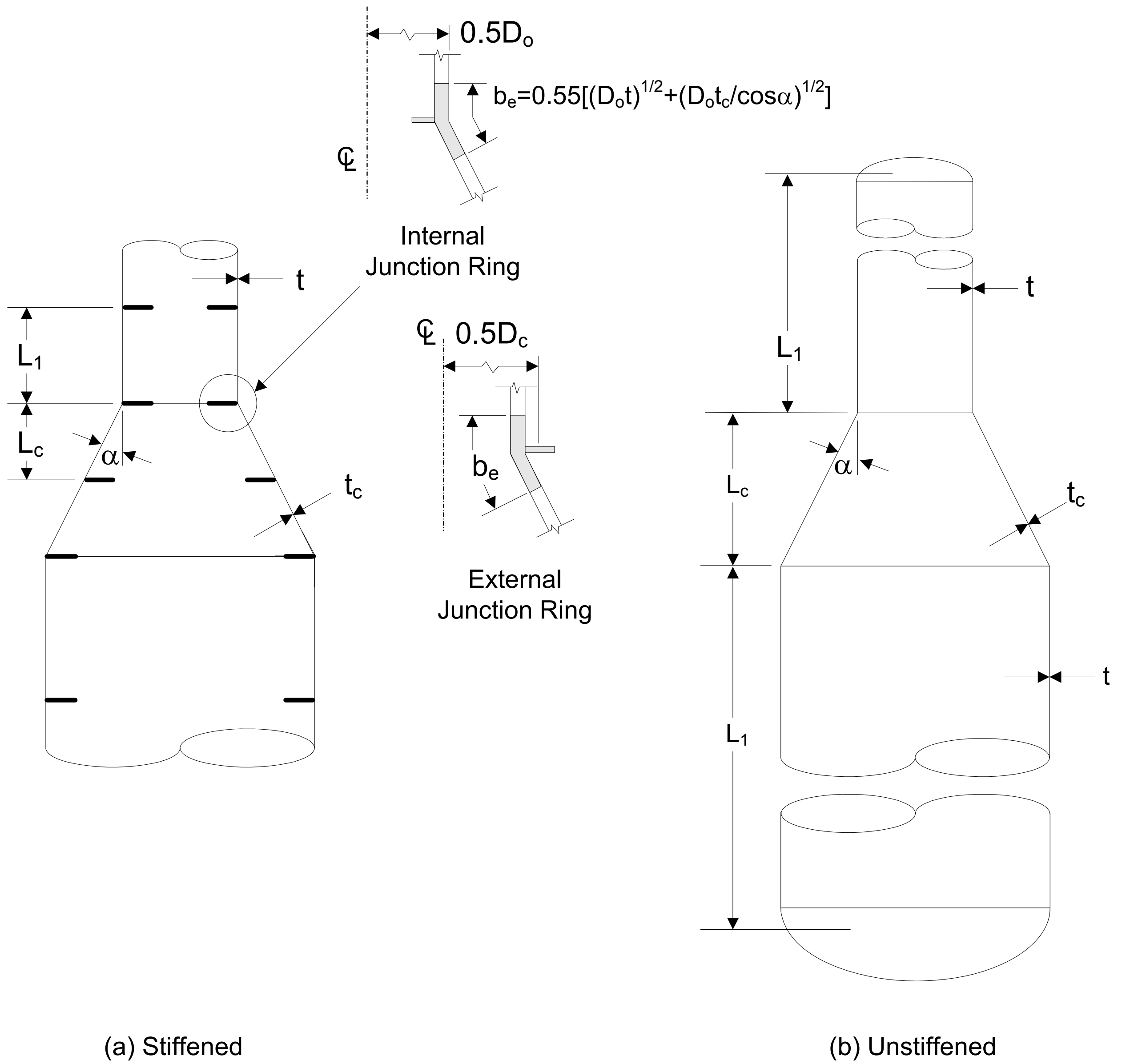
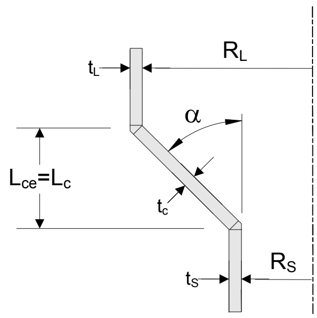
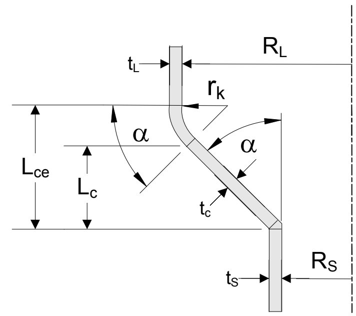


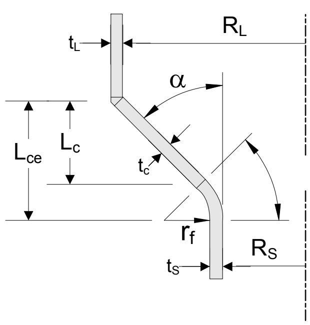
Figure 4.4.6 – Lines of Support or Unsupported Length for Unstiffened and Stiffened Conical Shells



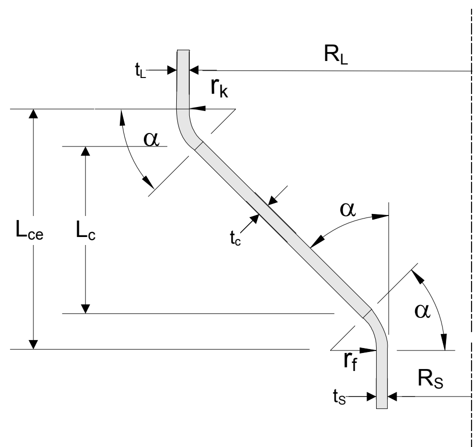
(a) Cone without a Knuckle at Large End without a Flare at the Small End



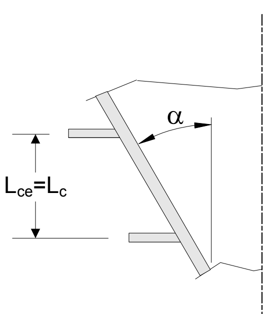
(b) Cone with a Knuckle at Large End without a Flare at the Small End



(c) Cone without a Knuckle at Large End with a Flare at the Small End



(d) Cone with a Knuckle at Large End with a Flare at the Small End



(e) Cone with Stiffening Rings

Figure 4.4.7 – Lines of Support or Unsupported Length for Unstiffened and Stiffened Conical Shell Transitions with or without a Knuckle

4.5 Design Rules for Openings in Shells and Heads

4.5.1 Scope

The rules in paragraph 4.5 are applicable for the design of nozzles in shells and heads subjected to internal pressure, external pressure, and external forces and moments from supplemental loads as defined in paragraph 4.1. Configurations, including dimensions and shape, and/or loading conditions that do not satisfy the rules of this paragraph 4.5 may be designed in accordance with Part 5.

4.5.2 Dimensions and Shape of Nozzles

4.5.2.1 Nozzles shall be circular, elliptical, or of any other shape which results from the intersection of a circular or elliptical cylinder with vessels of the shapes for which design equations are provided in paragraphs 4.3 and 4.4. The design rules in this paragraph shall only be used if the ratio of the inside diameter of the shell and the shell thickness is less than or equal to 400. In addition, the ratio of the diameter along the major axis to the diameter along the minor axis of the finished nozzle opening shall be less than or equal to 1.5.

4.5.2.2 Nozzle openings that do not satisfy the criteria of paragraph 4.5.2.1 and other geometries shall be designed in accordance with Part 5.

4.5.3 Method of Nozzle Attachment

4.5.3.1 Nozzles may be attached to the shell or head of a vessel by the following methods.

a) Welded Connections – Nozzles attachment by welding shall be in accordance with the requirements of paragraph 4.2.2. If other details not included in this paragraph are required, the nozzle detail shall be designed using Part 5.

b) Studded Connections – Nozzles may be made by means of studded pad type connections. The vessel shall have a flat surface machined on the shell, or on a built-up pad, or on a properly attached plate or fitting. Drilled holes to be tapped shall not penetrate within one-fourth of the wall thickness from the inside surface of the vessel after deducting corrosion allowance, unless at least the minimum thickness required as above is maintained by adding metal to the inside surface of the vessel. Where tapped holes are provided for studs, the threads shall be full and clean and shall engage the stud for a length, L_{st}, defined by the following equations.

$$L_{st} = \min\left[L_{st1},\ 1.5d_{st}\right] \tag{4.5.1}$$

where

$$L_{st1} = \max\left[d_{st},\ 0.75\left(\frac{S_{st}}{S_{tp}}\right)\right] \tag{4.5.2}$$

c) Threaded Connections – Pipes, tubes, and other threaded connections that conform to the ANSI/ASME Standard for Pipe Threads, General Purpose, Inch (ASME B1.20.1) may be screwed into a threaded hole in a vessel wall, provided the connection size is less than or equal to DN 50 (NPS 2) and the pipe engages the minimum number of threads specified in Table 4.5.1 after allowance has been made for curvature of the vessel wall. The thread shall be a standard taper pipe thread except that a straight thread of at least equal strength may be used if other sealing means to prevent leakage are provided. A built-up pad or a properly attached plate or fitting may be used to provide the metal thickness and number of threads required in Table 4.5.1, or to furnish reinforcement when required.

d) Expanded Connections – A pipe, tube, or forging may be attached to the wall of a vessel by inserting through an unreinforced opening and expanding into the shell, provided the diameter is not greater than DN 50 (NPS 2) pipe size. A pipe, tube, or forging not exceeding 150 mm (6 in.) in outside diameter may be attached to the wall of a vessel by inserting through a reinforced opening and expanding into the shell. The expanded connection shall be made using one of the following methods:

1) Firmly rolled in and beaded

2) Rolled in, beaded, and seal-welded around the edge of the bead

3) Expanded and flared not less than 3 mm (0.125 in.) over the diameter of the hole

4) Rolled, flared, and welded; or

5) Rolled and welded without flaring or beading, provided the ends extend at least 6 mm (0.25 in.), but no more than 10 mm (0.375 in.), through the shell and the throat of the weld is at least 5 mm (0.1875 in.), but no more than 8 mm (0.3125 in.).

4.5.3.2 Additional requirements for nozzle connections are as follows.

a) When the tube or pipe does not exceed 38 mm (1.5 in.) in outside diameter, the shell may be chamfered or recessed to a depth at least equal to the thickness of the tube or pipe and the tube or pipe may be rolled into place and welded. In no case shall the end of the tube or pipe extend more than 10 mm (0.375 in.) beyond the inside diameter of the shell.

b) Grooving of shell openings in which tubes and pipe are to be rolled or expanded is permissible.

c) Expanded connections shall not be used as a method of attachment to vessels used for the processing or storage of flammable and/or noxious gases and liquids unless the connections are seal-welded.

d) Reinforcing plates and saddles attached to the outside of a vessel shall be provided with at least one vent hole that may be tapped for a preliminary compressed air and soap solution (or equivalent) test for tightness of welds that seal off the inside of the vessel. These vent holes may be left open or may be plugged when the vessel is in service. If the holes are plugged, the plugging material used shall not be capable of sustaining pressure between the reinforcing plate and the vessel wall. Vent holes shall not be plugged during heat treatment.

4.5.4 Nozzle Neck Minimum Thickness Requirements

4.5.4.1 The minimum nozzle neck thickness for nozzles excluding access openings and openings for inspection shall be determined for internal and external pressure using paragraphs 4.3 and 4.4, as applicable. Corrosion allowance and the effects of external forces and moments from supplemental loads shall be considered in these calculations. The resulting nozzle neck thickness shall not be less than the smaller of the shell thickness or the thickness given in Table 4.5.2. Corrosion allowance shall be added to the minimum nozzle neck thickness.

4.5.4.2 The minimum nozzle neck thickness for access openings and openings for inspection shall be determined for internal and external pressure using paragraphs 4.3 and 4.4. Corrosion allowance shall be considered in these calculations.

4.5.5 Radial Nozzle in a Cylindrical Shell

4.5.5.1 The procedure to design a radial nozzle in a cylindrical shell subject to pressure loading is shown below. The parameters used in this design procedure are shown in Figures 4.5.1, 4.5.2, and 4.5.3.

a) STEP 1 – Calculate the limit of reinforcement along the vessel wall:

1) For integrally reinforced nozzles:

$$L_R = \min\left[\sqrt{R_{eff}t}, 2R_n\right] \tag{4.5.3}$$

2) For nozzles with reinforcing pads:

$$L_{R1} = \sqrt{R_{eff} t} + W \tag{4.5.4}$$

$$L_{R2} = \sqrt{\left(R_{eff} + t\right)\left(t + t_e\right)} \tag{4.5.5}$$

$$L_{R3} = 2R_n \tag{4.5.6}$$

$$L_R = \min\left[L_{R1}, L_{R2}, L_{R3}\right] \tag{4.5.7}$$

b) STEP 2 – Calculate the limit of reinforcement along the nozzle wall projecting outside the vessel surface:

$$L_{H1} = t + t_e + \sqrt{R_n t_n} \tag{4.5.8}$$

$$L_{H2} = L_{pr1} + t \qquad \textit{for nozzles inserted through the vessel wall} \tag{4.5.9}$$

$$L_{H2} = L_{pr1} \qquad \textit{for nozzles abutting the vessel wall} \tag{4.5.10}$$

$$L_{H3} = 8\left(t + t_e\right) \tag{4.5.11}$$

$$L_H = \min\left[L_{H1},\ L_{H2},\ L_{H3}\right] \tag{4.5.12}$$

c) STEP 3 – Calculate the limit of reinforcement along the nozzle wall projecting inside the vessel surface, if applicable:

$$L_{I1} = \sqrt{R_n t_n} \tag{4.5.13}$$

$$L_{I2} = L_{pr2} \tag{4.5.14}$$

$$L_{I3} = 8\left(t + t_e\right) \tag{4.5.15}$$

$$L_I = \min\left[L_{I1},\ L_{I2},\ L_{I3}\right] \tag{4.5.16}$$

d) STEP 4 – Determine the total available area near the nozzle opening (see Figure 4.5.1).

$$A_T = A_1 + f_{rn}(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{rp} A_5 \tag{4.5.17}$$

$$A_1 = \left(t L_R\right) \cdot \max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right] \tag{4.5.18}$$

$$\lambda = \min\left[\left\{\frac{\left(2R_n + t_n\right)}{\sqrt{\left(D_i + t_{eff}\right) t_{eff}}}\right\},\ 12.0\right] \tag{4.5.19}$$

$$A_2 = t_n\left(L_{pr3}+t\right)+0.78\left(\frac{t_{n2}^2}{t_n}\right)\sqrt{R_n t_{n2}} \quad \text{for Variable Thickness Openings where, } L_H > L_{pr3}+t \tag{4.5.20}$$

$$A_2 = t_n L_H \quad \text{for Variable Thickness Openings where, } L_H \le L_{pr3}+t \text{ or, for Uniform Thickness Openings} \tag{4.5.21}$$

$$A_3 = t_n L_I \tag{4.5.22}$$

$$A_{41} = 0.5L_{41}^2 \tag{4.5.23}$$

$$A_{42} = 0.5L_{42}^2 \tag{4.5.24}$$

$$A_{43} = 0.5L_{43}^2 \tag{4.5.25}$$

$$A_{5a} = Wt_e \tag{4.5.26}$$

$$A_{5b} = L_R t_e \quad \text{for nozzles inserted through the vessel wall} \tag{4.5.27}$$

$$A_{5b} = \left(L_R - t_n\right)t_e \quad \text{for nozzles abutting the vessel wall} \tag{4.5.28}$$

$$A_5 = \min\left[A_{5a},\ A_{5b}\right] \tag{4.5.29}$$

$$f_{rn} = \frac{S_n}{S} \tag{4.5.30}$$

$$f_{rp} = \frac{S_p}{S} \tag{4.5.31}$$

e) STEP 5 – Determine the effective radius of the shell as follows:

1) For cylindrical shells:

$$R_{eff} = 0.5D_i \tag{4.5.32}$$

2) For conical shells R_{eff} is the inside radius of the conical shell at the nozzle centerline to cone junction. The radius is measured normal to the longitudinal axis of the conical shell

f) STEP 6 – Determine the effective shell thickness for nozzles in cylindrical or conical shells as follows:

$$t_{eff} = t\left(\frac{tL_R + A_5 f_{rp}}{tL_R}\right) \tag{4.5.33}$$

g) STEP 7 – Determine the applicable forces:

$$f_N = PR_{xn}\left(L_H - t\right) \tag{4.5.34}$$

$$f_S = PR_{xs}\left(L_R + t_n\right) \tag{4.5.35}$$

$$f_Y = PR_{xs}R_{nc} \tag{4.5.36}$$

$$R_{xn} = \frac{t_n}{\ln\left[\dfrac{R_n + t_n}{R_n}\right]} \tag{4.5.37}$$

$$R_{xs} = \frac{t_{eff}}{\ln\left[\dfrac{R_{eff} + t_{eff}}{R_{eff}}\right]} \tag{4.5.38}$$

h) STEP 8 – Determine the average local primary membrane stress and the general primary membrane stress in the vessel:

$$\sigma_{avg} = \frac{\left(f_N + f_S + f_Y\right)}{A_T} \tag{4.5.39}$$

$$\sigma_{circ} = \frac{PR_{xs}}{t_{eff}} \tag{4.5.40}$$

i) STEP 9 – Determine the maximum local primary membrane stress at the nozzle intersection:

$$P_L = \max\left[\left(2\sigma_{avg} - \sigma_{circ}\right),\ \sigma_{circ}\right] \tag{4.5.41}$$

j) STEP 10 – The calculated maximum local primary membrane stress should satisfy Equation (4.5.42). If the nozzle is subjected to internal pressure, then the allowable stress, S_{allow}, is given by Equation (4.5.43). If the nozzle is subjected to external pressure, then the allowable stress is given by Equation (4.5.44) where F_{ha} is evaluated in paragraph 4.4 for the shell geometry being evaluated (e.g. cylinder, spherical shell, or formed head).

$$P_L \le S_{allow} \tag{4.5.42}$$

where,

$$S_{allow} = 1.5SE \qquad \textit{for internal pressure} \tag{4.5.43}$$

$$S_{allow} = F_{ha} \qquad \textit{for external pressure} \tag{4.5.44}$$

k) STEP 11 – Determine the maximum allowable working pressure of the nozzle:

$$P_{\max 1} = \frac{S_{allow}}{\dfrac{2A_p}{A_T} - \dfrac{R_{xs}}{t_{eff}}} \tag{4.5.45}$$

$$P_{\max 2} = S\left(\frac{t}{R_{xs}}\right) \tag{4.5.46}$$

$$P_{\max} = \min\left[P_{\max 1},\ P_{\max 2}\right] \tag{4.5.47}$$

where,

$$A_p = R_{xn}\left(L_H - t\right) + R_{xs}\left(L_R + t_n + R_{nc}\right) \tag{4.5.48}$$

4.5.5.2 If the nozzle is subject to external forces and moments from supplemental loads as defined in paragraph 4.1, then the local stresses at the nozzle-to-shell intersection shall be evaluated in accordance with paragraph 4.5.15.

4.5.6 Hillside Nozzle in a Cylindrical Shell

For a hillside nozzle in a cylindrical shell (see Figure 4.5.4), the design procedure in paragraph 4.5.5 shall be used with the following substitution.

$$R_{nc} = \max\left[\left(\frac{R_{ncl}}{2}\right),\ R_n\right] \tag{4.5.49}$$

where,

$$R_{ncl} = R_{eff}\left(\theta_1 - \theta_2\right) \tag{4.5.50}$$

$$\theta_1 = \cos^{-1}\left[\frac{D_X}{R_{eff}}\right] \tag{4.5.51}$$

$$\theta_2 = \cos^{-1}\left[\frac{D_X + R_n}{R_{eff}}\right] \tag{4.5.52}$$

4.5.7 Nozzle in a Cylindrical Shell Oriented at an Angle from the Longitudinal Axis

For a nozzle in a cylindrical shell oriented at an angle from the longitudinal axis, the design procedure in paragraph 4.5.5 shall be used with the following substitution (see Figure 4.5.5):

$$R_{nc} = \frac{R_n}{\sin\left[\theta\right]} \tag{4.5.53}$$

4.5.8 Radial Nozzle in a Conical Shell

For a radial nozzle in a conical shell (see Figure 4.5.6), the design procedure in paragraph 4.5.5 shall be used with the following substitutions.

$$f_S = \frac{P}{\cos[\alpha]}\left(R_{eff} + \frac{L_c}{2}\sin[\alpha]\right)\left(L_R + t_n\right) \tag{4.5.54}$$

$$f_Y = \frac{P\left(R_{eff} + \frac{R_{nc}}{2}\sin[\alpha]\right)R_{nc}}{\cos[\alpha]} \tag{4.5.55}$$

$$\sigma_{circ} = \frac{P\left(R_{eff} + L_c \sin[\alpha]\right)}{t_{eff}\cos[\alpha]} \tag{4.5.56}$$

$$P_{\max} = \frac{S_{allow}}{\frac{2A_p}{A_T} - \frac{R_{eff} + L_c\sin[\alpha]}{t_{eff}\cos[\alpha]}} \tag{4.5.57}$$

$$A_p = \left[R_{xn}\left(L_H - t\right) + \left(R_{eff} + \frac{L_c}{2}\sin[\alpha]\right)\left(\frac{L_R + t_n}{\cos[\alpha]}\right) + \frac{R_{nc}\left(R_{eff} + 0.5R_{nc}\sin[\alpha]\right)}{\cos[\alpha]}\right] \tag{4.5.58}$$

$$L_c = L_R + t_n + R_{nc} \tag{4.5.59}$$

4.5.9 Nozzle in a Conical Shell

4.5.9.1 If a nozzle in a conical shell is oriented perpendicular to the longitudinal axis (see Figure 4.5.7), then the design procedure in paragraph 4.5.8 shall be used with the following substitution.

$$R_{nc} = \frac{R_n}{\cos[\alpha]} \tag{4.5.60}$$

4.5.9.2 If a nozzle in a conical shell is oriented parallel to the longitudinal axis (see Figure 4.5.8), then the design procedure in paragraph 4.5.8 shall be used with the following substitution.

$$R_{nc} = \frac{R_n}{\sin[\alpha]} \tag{4.5.61}$$

4.5.10 Radial Nozzle in a Spherical Shell or Formed Head

4.5.10.1 The procedure to design a radial nozzle in a spherical shell or formed head subject to pressure loading is shown below. The parameters used in this design procedure are shown in Figure 4.5.9.

a) STEP 1 – Determine the effective radius of the shell or formed head as follows.

1) For spherical shells:

$$R_{eff} = 0.5D_i \tag{4.5.62}$$

2) For ellipsoidal heads:

$$R_{eff} = \frac{0.9D_i}{6}\left[2+\left(\frac{D_i}{2h}\right)^2\right] \tag{4.5.63}$$

3) For torispherical heads:

$$R_{eff} = L \tag{4.5.64}$$

b) STEP 2 – Calculate the limit of reinforcement along the vessel wall.

1) For integrally reinforced nozzles in spherical shells and ellipsoidal heads:

$$L_R = \min\left[\sqrt{R_{eff}t},\ 2R_n\right] \tag{4.5.65}$$

2) For integrally reinforced nozzles in torispherical heads:

$$L_{R1} = \frac{D_i}{2} - \left(D_R + R_n + t_n\right) \tag{4.5.66}$$

$$L_{R2} = \min\left[\sqrt{R_{eff}t},\ 2R_n\right] \tag{4.5.67}$$

$$L_R = \min\left[L_{R1},\ L_{R2}\right] \tag{4.5.68}$$

3) For pad reinforced nozzles:

$$L_{R1} = \sqrt{R_{eff}t} + W \tag{4.5.69}$$

$$L_{R2} = \sqrt{\left(R_{eff}+t\right)\left(t+t_e\right)} \tag{4.5.70}$$

$$L_{R3} = 2R_n \tag{4.5.71}$$

$$L_R = \min\left[L_{R1}, L_{R2}, L_{R3}\right] \tag{4.5.72}$$

c) STEP 3 – Calculate the limit of reinforcement along the nozzle wall projecting outside the vessel surface.

$$L_H = \min\left[t + t_e + F_p\sqrt{R_n t_n}, L_{pr1} + t\right] \quad \textit{for nozzles inserted through the vessel wall} \tag{4.5.73}$$

$$L_H = \min\left[t + t_e + F_p\sqrt{R_n t_n}, L_{pr1}\right] \quad \textit{for nozzles abutting the vessel wall} \tag{4.5.74}$$

For ellipsoidal and torispherical heads,

$$F_p = \min\left[C_n,\ C_p\right] \quad \textit{for } X_o > 0.35D_i \tag{4.5.75}$$

$$F_p = C_n \quad \textit{for } X_o \le 0.35D_i \tag{4.5.76}$$

$$C_p = \exp\left[\frac{0.35D_i - X_o}{16t}\right] \quad \textit{for ellipsoidal heads} \tag{4.5.77}$$

$$C_p = \exp\left[\frac{0.35D_i - X_o}{8t}\right] \quad \textit{for torispherical heads} \tag{4.5.78}$$

$$X_o = D_R + R_n + t_n \tag{4.5.79}$$

For spherical shells and heads,

$$F_p = C_n \tag{4.5.80}$$

The parameter C_n is given by Equation (4.5.81).

$$C_n = \min\left[\left(\frac{t + t_e}{t_n}\right)^{0.35}, 1.0\right] \tag{4.5.81}$$

d) STEP 4 – Calculate the limit of reinforcement along the nozzle wall projecting inside the vessel surface, if applicable.

$$L_I = \min\left[F_p\sqrt{R_n t_n}, L_{pr2}\right] \tag{4.5.82}$$

e) STEP 5 – Determine the total available area near the nozzle opening (see Figure 4.5.1) where f_{rn} and f_{rp} are given by Equations (4.5.30) and (4.5.31), respectively.

$$A_T = A_1 + f_{rn}(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{rp}A_5 \tag{4.5.83}$$

$$A_1 = tL_R \tag{4.5.84}$$

$$A_2 = t_n\left(L_{pr3} + t\right) + 0.78\left(\frac{t_{n2}^2}{t_n}\right)\sqrt{R_n t_{n2}} \qquad \begin{array}{l} for\ Variable\,Thickness\,Openings \\ where,\ L_H > L_{pr3} + t \end{array} \tag{4.5.85}$$

$$A_2 = t_n L_H \qquad \begin{array}{l} for\,Variable\,Thickness\,Openings \\ where,\ L_H \le L_{pr3} + t \\ or,\ for\,Uniform\,Thickness\,Openings \end{array} \tag{4.5.86}$$

$$A_3 = t_n L_I \tag{4.5.87}$$

$$A_{41} = 0.5L_{41}^2 \tag{4.5.88}$$

$$A_{42} = 0.5L_{42}^2 \tag{4.5.89}$$

$$A_{43} = 0.5L_{43}^2 \tag{4.5.90}$$

$$A_{5a} = Wt_e \tag{4.5.91}$$

$$A_{5b} = L_R t_e \qquad for\,nozzles\,inserted\,through\,the\,vessel\,wall \tag{4.5.92}$$

$$A_{5b} = \left(L_R - t_n\right)t_e \qquad for\,nozzles\,abutting\,the\,vessel\,wall \tag{4.5.93}$$

$$A_5 = \min\left[A_{5a},\ A_{5b}\right] \tag{4.5.94}$$

f) STEP 6 – Determine the applicable forces.

$$f_N = PR_{xn}\left(L_H - t\right) \tag{4.5.95}$$

$$f_S = \frac{PR_{xs}\left(L_R + t_n\right)}{2} \tag{4.5.96}$$

$$f_Y = \frac{PR_{xs}R_{nc}}{2} \tag{4.5.97}$$

$$R_{xn} = \frac{t_n}{\ln\left[\dfrac{R_n + t_n}{R_n}\right]} \tag{4.5.98}$$

$$R_{xs} = \frac{t_{eff}}{\ln\left[\dfrac{R_{eff} + t_{eff}}{R_{eff}}\right]} \tag{4.5.99}$$

g) STEP 7 – Determine the effective thickness for nozzles in spherical shells or ellipsoidal or torispherical heads as follows.

$$t_{eff} = t\left(\frac{tL_R + A_5 f_{rp}}{tL_R}\right) \tag{4.5.100}$$

h) STEP 8 – Determine the average local primary membrane stress and the general primary membrane stress in the vessel.

$$\sigma_{avg} = \frac{\left(f_N + f_S + f_Y\right)}{A_T} \tag{4.5.101}$$

$$\sigma_{circ} = \frac{PR_{xs}}{2t_{eff}} \tag{4.5.102}$$

i) STEP 9 – Determine the maximum local primary membrane stress at the nozzle intersection.

$$P_L = \max\left[\left\{2\sigma_{avg} - \sigma_{circ}\right\},\ \sigma_{circ}\right] \tag{4.5.103}$$

j) STEP 10 – The calculated maximum local primary membrane stress should satisfy Equation (4.5.104). If the nozzle is subjected to internal pressure, then the allowable stress, S_{allow}, is given by Equation (4.5.43). If the nozzle is subjected to external pressure, then the allowable stress is given by Equation (4.5.44).

$$P_L \le S_{allow} \tag{4.5.104}$$

k) STEP 11 – Determine the maximum allowable working pressure of the nozzle.

$$P_{\max 1} = \frac{S_{allow}}{\dfrac{2A_p}{A_T} - \dfrac{R_{xs}}{2t_{eff}}} \tag{4.5.105}$$

$$P_{\max 2} = 2S\left(\frac{t}{R_{xs}}\right) \tag{4.5.106}$$

$$P_{\max} = \min\left[P_{\max 1},\ P_{\max 2}\right] \tag{4.5.107}$$

where,

$$A_p = R_{xn}\left(L_H - t\right) + \frac{R_{xs}\left(L_R + t_n + R_{nc}\right)}{2} \tag{4.5.108}$$

4.5.10.2 If the nozzle is subject to external forces and moments from supplemental loads as defined in paragraph 4.1, then the local stresses at the nozzle-to-shell intersection shall be evaluated in accordance with paragraph 4.5.15.

4.5.11 Hillside or Perpendicular Nozzle in a Formed Head

4.5.11.1 If a hillside or perpendicular nozzle is located in an ellipsoidal head (see Figure 4.5.10), the design procedure in paragraph 4.5.10 shall be used with the following substitution.

$$D_R = \frac{D_i}{2}\sqrt{1-\frac{D_T^2}{h^2}} \tag{4.5.109}$$

4.5.11.2 If a hillside or perpendicular nozzle is located in a head type that is not an ellipsoidal heads (see Figure 4.5.10), the design procedure in paragraph 4.5.10 shall be used with the following substitution.

$$R_{nc} = R_{eff}\left(\theta_1 - \theta_2\right) \tag{4.5.110}$$

$$\theta_1 = \cos^{-1}\left[\frac{D_R}{R_{eff}}\right] \tag{4.5.111}$$

$$\theta_2 = \cos^{-1}\left[\frac{D_R + R_n}{R_{eff}}\right] \tag{4.5.112}$$

4.5.12 Circular Nozzles in a Flat Head

4.5.12.1 The procedure to design a nozzle in a flat head subject to pressure loading is shown below. The parameters used in this design procedure are shown in Figures 4.5.9. As an alternative, a central nozzle in an integral flat head may be designed using the procedure in paragraph 4.6.3.

a) STEP 1 – Calculate the maximum unit moment at the nozzle intersection.

$$M_o = \frac{St_{rf}^4}{6\left(t + C_p t_e\right)^2} \tag{4.5.113}$$

$$C_p = \min\left[\left\{\frac{(w+0.5L_{42})t_e}{R_n t}\right\},\ 0.6\right] \tag{4.5.114}$$

b) STEP 2 – Calculate the nozzle parameters.

$$\lambda_n = \frac{1.285}{\sqrt{R_{nm}t_n}} \tag{4.5.115}$$

$$C_1 = \sinh^2\left[C_L\right] + \sin^2\left[C_L\right] \tag{4.5.116}$$

$$C_2 = \sinh^2\left[C_L\right] - \sin^2\left[C_L\right] \tag{4.5.117}$$

$$C_L = \min\left[\left\{\lambda_n\left(L_{pr1} + t + L_{pr2}\right)\right\},\ 6.0\right] \tag{4.5.118}$$

$$C_3 = \frac{L_{pr1} + t}{L_{pr1} + t + \min\left[\left(\lambda_n\right)^{-1},\ L_{pr2}\right]} \tag{4.5.119}$$

$$R_{nm} = R_n + 0.5t_n \tag{4.5.120}$$

$$R_{xn} = \frac{t_n}{\ln\left[\dfrac{R_n + t_n}{R_n}\right]} \tag{4.5.121}$$

$$x_t = 0.5\lambda_n\left(t + t_e + L_{41} + L_{43}\right) \quad \textit{for nozzles inserted through the vessel wall} \tag{4.5.122}$$

$$x_t = 0.5\lambda_n\left(t_e + L_{41}\right) \quad \textit{for nozzles abutting the vessel wall} \tag{4.5.123}$$

$$C_t = \exp\left[-x_t\right] \tag{4.5.124}$$

c) STEP 3 – Determine the maximum local primary membrane stress in the nozzle at the intersection.

$$P_L = \frac{2M_o\lambda_n^2 R_{nm} C_t C_1 C_3}{t_n C_2} + \frac{PR_{xn}}{t_n} \tag{4.5.125}$$

d) STEP 4 – The maximum local primary membrane stress at the nozzle intersection shall satisfy Equation (4.5.126). If the nozzle is subjected to internal pressure, then the allowable stress, S_{allow}, is given by Equation (4.5.43). If the nozzle is subjected to external pressure, then the allowable stress is given by Equation (4.5.44).

$$P_L \le S_{allow} \tag{4.5.126}$$

4.5.12.2 If the nozzle is subject to external forces and moments from supplemental loads as defined in paragraph 4.1, then the local stresses at the nozzle-to-shell intersection shall be evaluated in accordance with paragraph 4.5.15.

4.5.13 Spacing Requirements for Nozzles

4.5.13.1 If the limits of reinforcement determined in accordance with paragraph 4.5.5 for nozzles in cylindrical or conical shells or paragraph 4.5.10 for nozzles in spherical or formed heads, do not overlap, no additional analysis is required. If the limits of reinforcement overlap, the following procedure shall be used or the design shall be evaluated in accordance with the design by analysis rules of Part 5.

4.5.13.2 The maximum local primary membrane stress and the nozzle maximum allowable working pressure shall be determined following paragraphs 4.5.5 or 4.5.10, for each individual nozzle with the value of L_R determined as follows.

a) For two openings with overlapping limits of reinforcement (see Figure 4.5.11):

$$L_R = L_S\left(\frac{R_{na}}{R_{na}+R_{nb}}\right) \qquad \textit{for nozzle A} \tag{4.5.127}$$

$$L_R = L_S\left(\frac{R_{nb}}{R_{na}+R_{nb}}\right) \qquad \textit{for nozzle B} \tag{4.5.128}$$

b) For three openings with overlapping limits of reinforcement (see Figure 4.5.12):

$$L_R = \min\left[L_{S1}\left(\frac{R_{na}}{R_{na}+R_{nb}}\right),\ L_{S2}\left(\frac{R_{na}}{R_{na}+R_{nc}}\right)\right] \qquad \textit{for nozzle A} \tag{4.5.129}$$

$$L_R = \min\left[L_{S1}\left(\frac{R_{nb}}{R_{na}+R_{nb}}\right),\ L_{S3}\left(\frac{R_{nb}}{R_{nb}+R_{nc}}\right)\right] \qquad \textit{for nozzle B} \tag{4.5.130}$$

$$L_R = \min\left[L_{S2}\left(\frac{R_{nc}}{R_{na}+R_{nc}}\right),\ L_{S3}\left(\frac{R_{nc}}{R_{nb}+R_{nc}}\right)\right] \qquad \textit{for nozzle C} \tag{4.5.131}$$

c) For more than three openings with overlapping limits of reinforcement, repeat the above procedure for each pair of adjacent nozzles.

4.5.14 Strength of Nozzle Attachment Welds

4.5.14.1 The strength of nozzle attachment welds shall be sufficient to resist the discontinuity force imposed by pressure for nozzles attached to a cylindrical, conical, or spherical shell or formed head as determined in paragraph 4.5.14.2. Nozzles attached to flat heads shall have their strength of attachment welds evaluated as determined in paragraph 4.5.14.3. The effects of external forces and moments from supplemental loads shall be considered.

4.5.14.2 The procedure to evaluate attachment welds of nozzles in a cylindrical, conical, or spherical shell or formed head subject to pressure loading is shown below.

a) STEP 1 – Determine the discontinuity force factor

1) For nozzles abutting the vessel wall:

$$k_y = 1.0 \tag{4.5.132}$$

2) For nozzles inserted through the vessel wall:

$$k_y = \frac{R_{nc}+t_n}{R_{nc}} \tag{4.5.133}$$

b) STEP 2 – Calculate Weld Length Resisting Discontinuity Force

1) Weld length of nozzle to shell weld

$$L_\tau = \frac{\pi}{2}\left(R_n + t_n\right) \qquad \textit{for radial nozzles} \tag{4.5.134}$$

$$L_{\tau} = \frac{\pi}{2}\sqrt{\frac{\left(R_{nc}+t_n\right)^2+\left(R_n+t_n\right)^2}{2}} \qquad \textit{for nonradial nozzles} \tag{4.5.135}$$

2) Weld length of pad to shell weld

$$L_{\tau p} = \frac{\pi}{2}\left(R_n+t_n+W\right) \qquad \textit{for radial nozzles} \tag{4.5.136}$$

$$L_{\tau p} = \frac{\pi}{2}\sqrt{\frac{\left(R_{nc}+t_n+W\right)^2+\left(R_n+t_n+W\right)^2}{2}} \qquad \textit{for nonradial nozzles} \tag{4.5.137}$$

c) STEP 3 – Compute the weld throat dimensions, as applicable.

$$L_{41T} = 0.7071 L_{41} \tag{4.5.138}$$

$$L_{42T} = 0.7071 L_{42} \tag{4.5.139}$$

$$L_{43T} = 0.7071 L_{43} \tag{4.5.140}$$

d) STEP 4 – Determine if the weld sizes are acceptable.

1) If the nozzle is integrally reinforced, and the computed shear stress in the weld given by Equation (4.5.141) satisfies Equation (4.5.142), then the design is complete. If the shear stress in the weld does not satisfy Equation (4.5.142), increase the weld size and return to STEP 3. For nozzles on heads, A_2 and A_3 are to be calculated using $F_p = 1.0$, when computing f_{welds} using Equation (4.5.143).

$$\tau = \frac{f_{welds}}{L_{\tau}\left(0.49L_{41T}+0.6t_{w1}+0.49L_{43T}\right)} \tag{4.5.141}$$

$$\tau \le S \tag{4.5.142}$$

where,

$$f_{welds} = \min\left[f_Y k_y, 1.5S_n\left(A_2+A_3\right)\right] \tag{4.5.143}$$

2) If the nozzle is pad reinforced, and the computed shear stresses in the welds given by Equations (4.5.144) through (4.5.146) satisfy Equation (4.5.147), then the design is complete. If the shear stress in the weld does not satisfy Equation (4.5.147), increase the weld size and return to STEP 3.

$$\tau_1 = \frac{f_{ws}}{L_{\tau}\left(0.6t_{w1}+0.49L_{43T}\right)} \tag{4.5.144}$$

$$\tau_2 = \frac{f_{wp}}{L_{\tau}\left(0.6t_{w2}+0.49L_{41T}\right)} \tag{4.5.145}$$

$$\tau_3 = \frac{f_{wp}}{L_{\tau p}\left(0.49L_{42T}\right)} \tag{4.5.146}$$

$$\max\left[\tau_1,\ \tau_2,\ \tau_3\right] \le S \tag{4.5.147}$$

where,

$$f_{ws} = \frac{f_{welds} k_y t \cdot S}{t \cdot S + t_e S_p} \tag{4.5.148}$$

$$f_{wp} = \frac{f_{welds} k_y t_e S_p}{t \cdot S + t_e S_p} \tag{4.5.149}$$

3) If the nozzle is pad reinforced, and the computed shear stress in the nozzle wall given by Equation (4.5.150) satisfies Equation (4.5.151), then the design is complete. If the shear stress in the nozzle wall does not satisfy Equation (4.5.151), increase the nozzle thickness in accordance with paragraph 4.5.4.

$$\tau_n = \frac{\left(P_L - \dfrac{PR_n}{t_n}\right)t_e}{1.4t_n} \tag{4.5.150}$$

$$\tau_n \le 1.5S_n \tag{4.5.151}$$

4.5.14.3 The procedure to evaluate attachment welds of a nozzle in a flat head subject to pressure loading is shown below.

a) STEP 1 – Compute the weld throat dimensions, as applicable.

$$L_{41T} = 0.7071L_{41} \tag{4.5.152}$$

$$L_{42T} = 0.7071L_{42} \tag{4.5.153}$$

$$L_{43T} = 0.7071L_{43} \tag{4.5.154}$$

b) STEP 2 – Determine if the weld sizes are acceptable.

1) If the nozzle is integrally reinforced and inserted through the flat head, and the computed shear stress in the welds given by Equations (4.5.155) and (4.5.156) satisfy Equation (4.5.157), then the design is complete. If the shear stress in the welds does not satisfy Equation (4.5.157), increase the weld size and return to STEP 1.

$$\tau_1 = \frac{V_s}{0.6t_{x1} + 0.49L_{43T}} \tag{4.5.155}$$

$$\tau_2 = \frac{V_s}{0.6t_{x2} + 0.49L_{41T}} \tag{4.5.156}$$

$$\max\left[\tau_1,\ \tau_2\right] \le S \tag{4.5.157}$$

where

$$V_s = \frac{0.3St_{rf}^4}{t^3} \tag{4.5.158}$$

$$t_{x1} = \min\left[t_{w1},\ 0.5t\right] \tag{4.5.159}$$

$$t_{x2} = \min\left[\max\left[\left(t_{w1} - 0.5t\right),\ 0\right],\ 0.5t\right] \tag{4.5.160}$$

2) If the nozzle is pad reinforced and inserted through the flat head, and the computed shear stress in the welds given by Equations (4.5.161) through (4.5.163) satisfy Equation (4.5.164), then the design is complete. If the shear stress in the welds does not satisfy Equation (4.5.164), increase the weld size and return to STEP 1.

$$\tau_1 = \frac{V_s}{0.6t_{w1} + 0.49L_{43T}} \tag{4.5.161}$$

$$\tau_2 = \frac{V_s}{0.6t_{w2} + 0.49L_{41T}} \tag{4.5.162}$$

$$\tau_3 = \frac{V_s\left(R_n + t_n\right)}{0.49L_{42T}(R_n + t_n + W)} \tag{4.5.163}$$

$$\max\left[\tau_1,\ \tau_2,\ \tau_3\right] \le S \tag{4.5.164}$$

The parameter V_s is given by Equation (4.5.158).

3) If the nozzle is integrally reinforced and abutting the flat head, and the computed shear stress in the weld given by Equation (4.5.165) satisfies Equation (4.5.166), then the design is complete. If the shear stress in the weld does not satisfy Equation (4.5.166), increase the weld size and return to STEP 1.

$$\tau = \frac{2M_o}{t\left(0.6t_{w1} + 0.49L_{41T}\right)} \tag{4.5.165}$$

$$\tau \le S \tag{4.5.166}$$

4.5.15 Local Stresses in Nozzles in Shells and Formed Heads from External Loads

Localized stresses at nozzle locations in shells and formed heads shall be evaluated using one of the method shown below. For each method, the acceptance criteria shall be in accordance with Part 5.

a) Nozzles in cylindrical shells – stress calculations shall be in accordance with WRC 107 or WRC 297.

b) Nozzles in formed heads – stress calculations shall be in accordance with WRC 107.

c) For all configurations, and as an alternative to paragraph 4.5.15.a and paragraph 4.5.15.b, the stress calculations may be performed using a numerical analysis such as the finite element method.

4.5.16 Inspection Openings

4.5.16.1 All pressure vessels for use with compressed air and those subject to internal corrosion or having parts subject to erosion or mechanical abrasion (see paragraph 4.1.4), except as permitted otherwise in this paragraph, shall be provided with a suitable manhole, handhole, or other inspection opening(s) for examination and cleaning. Compressed air as used in this paragraph is not intended to include air which has had moisture removed to provide an atmospheric dew point of -46°C (-50°F) or less.

4.5.16.2 Inspection openings maybe omitted in the shell side of fixed tubesheet heat exchangers. When inspection openings are not provided, the Manufacturer's Data Report shall include one of the following notations under remarks:

a) “Paragraph 4.5.16.2” when inspection openings are omitted in fixed tubesheet heat exchangers;

b) “Paragraph 4.5.16.3”, “Paragraph 4.5.16.4”, “Paragraph 4.5.16.5” when provision for inspection is made in accordance with one of these paragraphs;

c) The statement "for noncorrosive service.”

4.5.16.3 Vessels over 300 mm (12 in.) inside diameter under air pressure which also contain, as an inherent requirement of their operation, other substances which will prevent corrosion need not have openings for inspection only, provided the vessel contains suitable openings through which inspection can be made conveniently, and provided such openings are equivalent in size and number to the requirements for inspection openings in paragraph 4.5.16.6.

4.5.16.4 For vessels 300 mm (12 in.) or less in inside diameter, openings for inspection only may be omitted if there are at least two removable pipe connections not less than DN 20 (NPS ¾).

4.5.16.5 Vessels less than 400 mm (16 in.) and over 300 mm (12 in.) inside diameter shall have at least two handholes or two threaded pipe plug inspection openings of not less than DN 40 (NPS 1-1/2) except as permitted by the following: when vessels less than 400 mm (16 in.) and over 300 mm (12 in.) inside diameter are to be installed so that inspection cannot be made without removing the vessel from the assembly, openings for inspection only may be omitted provided there are at least two removable pipe connections of not less than DN 40 (NPS 1-1/2).

4.5.16.6 Vessels that require access or inspection openings shall be equipped as follows:

a) All vessels less than 450 mm (18 in.) and over 300 mm (12 in.) inside diameter shall have at least two handholes or two plugged, threaded inspection openings of not less than DN 40 (NPS 1-1/2);

b) All vessels 450 mm (18 in.) to 900 mm (36 in.), inclusive, inside diameter shall have a manhole or at least two handholes or two plugged, threaded inspection openings of not less than DN 50 (NPS 2);

c) All vessels over 900 mm (36 in.) inside diameter shall have a manhole, except that those whose shape or use makes one impracticable shall have at least two handholes 100 mm x 150 mm (4 in. x 6 in.) or two equal openings of equivalent area;

d) When handholes or pipe plug openings are permitted for inspection openings in place of a manhole, one handhole or one pipe plug opening shall be in each head or in the shell near each head service;

e) Openings with removable heads or cover plates intended for other purposes may be used in place of the required inspection openings provided they are equal at least to the size of the required inspection openings;

f) A single opening with removable head or cover plate may be used in place of all the smaller inspection openings provided it is of such size and location as to afford at least an equal view of the interior;

g) Flanged and/or threaded connections from which piping, instruments, or similar attachments can be removed may be used in place of the required inspection openings provided that:

    1) The connections are at least equal to the size of the required openings; and

2) The connections are sized and located to afford at least an equal view of the interior as the required inspection openings.

4.5.16.7 When inspection or access openings are required, they shall comply at least with the following requirements.

a) An elliptical or obround manhole shall be not less than 300 mm x 400 mm (12 in. x 16 in.). A circular manhole shall be not less than 400 mm (16 in.) inside diameter.

b) A handhole opening shall be not less than 50 mm x 75 mm (2 in. x 3 in.), but should be as large as is consistent with the size of the vessel and the location of the opening.

4.5.16.8 All access and inspection openings in a shell or unstayed head shall be designed in accordance with the rules of this Part for openings.

4.5.16.9 When a threaded opening is to be used for inspection or cleaning purposes, the closing plug or cap shall be of a material suitable for the pressure and no material shall be used at a temperature exceeding the maximum temperature allowed in Part 3 for that material. The thread shall be a standard taper pipe thread except that a straight thread of at least equal strength maybe used if other sealing means to prevent leakage are provided.

4.5.16.10 Manholes of the type in which the internal pressure forces the cover plate against a flat gasket shall have a minimum gasket bearing width of 17 mm (0.6875 in.).

4.5.17 Reinforcement of Openings Subject to Compressive Stress

4.5.17.1 The reinforcement for openings in cylindrical and conical vessels subject to compressive stress that do not exceed 25% of the cylinder diameter or 80% of the ring spacing into which the opening is placed may be designed in accordance with the following rules. Openings in cylindrical and conical vessels that exceed these limitations shall be designed in accordance with Part 5.

4.5.17.2 Reinforcement for nozzle openings in cylindrical and conical vessels designed for external pressure alone shall be in accordance with the requirements of paragraph 4.5.5 through paragraph 4.5.9, as applicable. The required thickness shall be determined in accordance with paragraph 4.5.4.

4.5.17.3 For cylindrical and conical vessels designed for axial compression (which includes axial load and/or bending moment) without external pressure, the reinforcement of openings shall be in accordance with the following:

$$A_r = 0 \qquad for \quad d \le 0.4\sqrt{Rt} \tag{4.5.167}$$

$$A_r = 0.5dt_r \qquad for \quad d > 0.4\sqrt{Rt} \quad and \quad \gamma_n \le \left(\frac{(R/t)}{291} + 0.22\right)^2 \tag{4.5.168}$$

$$A_r = dt_r \qquad for \quad d > 0.4\sqrt{Rt} \quad and \quad \gamma_n > \left(\frac{(R/t)}{291} + 0.22\right)^2 \tag{4.5.169}$$

where,

$$\gamma_n = \left(\frac{d}{2\sqrt{Rt}}\right) \tag{4.5.170}$$

4.5.17.4 The reinforcement shall be placed within a distance of $0.75\sqrt{Rt}$ from the edge of the opening. Reinforcement available from the nozzle neck shall be limited to a thickness not exceeding the shell plate thickness at the nozzle attachment, and be placed within a limit measured normal to the outside surface of the vessel shell of $0.5\sqrt{(d/2)t_n}$, but not exceeding $2.5t_n$.

4.5.17.5 For cylindrical and conical vessels designed for axial compression in combination with external pressure, the reinforcement shall be the larger of that required for external pressure alone, paragraph 4.5.17.2, or axial compression alone, paragraph 4.5.17.3. Required reinforcement shall be placed within the limits described in paragraph 4.5.17.4.

4.5.18 Nomenclature

A_1	area contributed by the vessel wall.
A_2	area contributed by the nozzle outside the vessel wall.
A_3	area contributed by the nozzle inside the vessel wall.
A_{41}	area contributed by the outside nozzle fillet weld.
A_{42}	area contributed by the pad to vessel fillet weld.
A_{43}	area contributed by the inside nozzle fillet weld.
A_5	area contributed by the reinforcing pad.
A_P	area resisting pressure, used to determine the nozzle opening discontinuity force.
A_r	area of reinforcement required.
A_T	total area within the assumed limits of reinforcement.
α	one-half of the apex angle of a conical shell.
D_i	inside diameter of a shell or head.
D_D	distance from a major structural discontinuity to the nozzle center line.
D_R	distance from the head center line to the nozzle center line.
D_T	distance from the head tangent line to the nozzle center line..
D_X	distance from the cylinder center line to the nozzle center line.
d	inside diameter of the opening.
d_{st}	nominal diameter of the stud.
E	weld joint factor (see paragraph 4.2); $E = 1.0$ if the nozzle does not intersect a weld seam.
E_{st}	engagement length of a stud.
F_p	nozzle attachment factor.
F_{ha}	minimum value of the allowable compressive stress of the shell and nozzle material from paragraph 4.4, evaluated at the design temperature.
f_N	force from internal pressure in the nozzle outside of the vessel.
f_{rn}	nozzle material factor.
f_{rp}	pad material factor.
f_S	force from internal pressure in the shell.
f_y	discontinuity force from pressure.

f_Y discontinuity force from internal pressure.

f_{ws} discontinuity force carried by welds t_{w1} and L_{43}.

f_{wn} discontinuity force carried by welds L_{41} and t_{w2} and weld L_{42}.

F_p nozzle attachment factor.

h height of the ellipsoidal head measured to the inside surface.

k_y discontinuity force factor that adjusts the discontinuity force to the nozzle outer diameter.

L_{41} weld leg length of the outside nozzle fillet weld.

L_{42} weld leg length of the pad to vessel fillet weld.

L_{43} weld leg length of the inside nozzle fillet weld.

L inside crown radius of a torispherical head.

L_H effective length of nozzle wall outside the vessel.

L_I effective length of nozzle wall inside the vessel.

L_{41T} throat dimension of the outside nozzle fillet weld.

L_{42T} throat dimension for the pad to vessel fillet weld.

L_{43T} throat dimension for inside nozzle fillet weld.

L_R effective length of the vessel wall.

L_{pr1} nozzle projection from the outside of the vessel wall.

L_{pr2} nozzle projection from the inside of the vessel wall.

L_{pr3} length of variable thickness, t_n, from the outside of the vessel wall.

L_τ weld length of the nozzle to shell weld.

$L_{\tau p}$ weld length of the pad to shell weld.

P internal or external design pressure.

$P_{\max}$ nozzle maximum allowable pressure.

P_L maximum local primary membrane stress at the nozzle intersection.

R vessel inside radius.

R_m vessel mean radius.

R_{eff} effective pressure radius.

R_n nozzle inside radius.

R_{nc} radius of the nozzle opening in the vessel along the long chord, for radial nozzles $R_{nc} = R_n$

R_{nm} nozzle mean radius.

R_{xn} nozzle radius for force calculation.

R_{xs} shell radius for force calculation.

S allowable stress from Annex 3.A for the vessel at the design temperature.

S_n allowable stress from Annex 3.A for the nozzle at the design temperature.

S_p allowable stress from Annex 3.A for the pad at the design temperature.

S_{st} allowable stress from Annex 3.A of the stud material at the design temperature.

S_{tp}	allowable stress from Annex 3.A of the tapped material at the design temperature.
σ_{avg}	average primary membrane stress.
σ_{circ}	general primary membrane stress.
θ	angle between the nozzle center line and the vessel center line.
t	nominal thickness of the vessel wall.
t_e	thickness of the reinforcing pad.
t_{eff}	effective thickness used in the calculation of pressure stress near the nozzle opening.
t_n	nominal thickness of the nozzle wall.
t_{n2}	nominal wall thickness of the thinner portion of a variable thickness nozzle.
t_r	thickness of shell required for axial compression loads without external pressure.
t_{rf}	minimum required flat head thickness, exclusive of corrosion allowance, as required by paragraph 4.6.
t_{w1}	nozzle to shell groove weld depth.
t_{w2}	nozzle to reinforcing pad groove weld depth.
τ	average “effective” shear stress in welds due to pressure (includes joint efficiency).
V_s	shear load.
W	width of the reinforcing pad.
X_o	distance from the nozzle outside diameter to the head center.

4.5.19 Tables

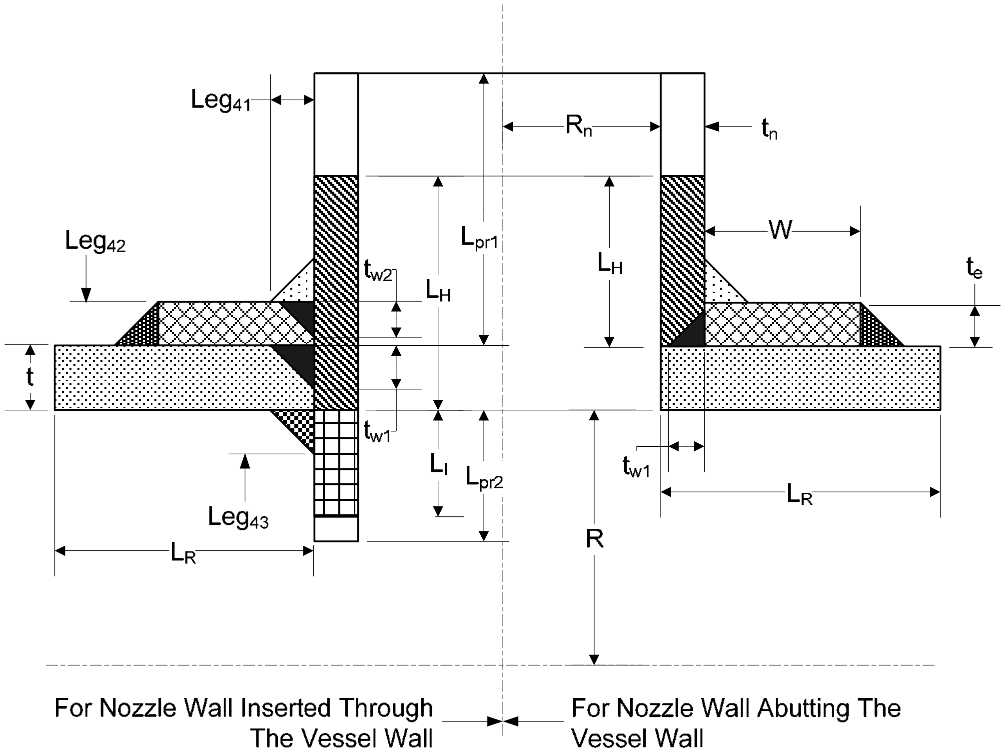
Table 4.5.1 – Minimum Number of Pipe Threads for Connections

Size Of Pipe	Threads Engaged	Minimum Plate Thickness Required
DIN 15, 20 (NPS 0.5, 0.75 in.)	6	11 mm (0.43 in.)
DIN 25, 32, 40 (NPS 1.0, 1.25, 1.5 in.)	7	16 mm (0.61 in.)
DIN 50 (NPS 2 in.)	8	18 mm (0.70 in.)

Table 4.5.2 – Nozzle Minimum Thickness Requirements

Nominal Size	Minimum Thickness	
	mm	in
DN 6 (NPS 1/8)	1.51	0.060
DN 8 (NPS 1/4)	1.96	0.077
DN 10 (NPS 3/8)	2.02	0.088
DN 15 (NPS 1/2)	2.42	0.095
DN 20 (NPS 3/4)	2.51	0.099
DN 25 (NPS 1)	2.96	0.116
DN 32 (NPS 1 1/4)	3.12	0.123
DN 40 (NPS 1 1/2)	3.22	0.127
DN 50 (NPS 2)	3.42	0.135
DN 65 (NPS 2 1/2)	4.52	0.178
DN 80 (NPS 3)	4.80	0.189
DN 90 (NPS 3 1/2)	5.02	0.198
DN 100 (NPS 4)	5.27	0.207
DN 125 (NPS 5)	5.73	0.226
DN 150 (NPS 6)	6.22	0.245
DN 200 (NPS 8)	7.16	0.282
DN 250 (NPS 10)	8.11	0.319
$\geq$ DN 300 (NPS 12)	8.34	0.328
Note: For nozzles having a specified outside diameter not equal to the outside diameter of an equivalent standard DN (NPS) size, the DN (NPS) chosen from the table shall be one having an equivalent outside diameter larger than the actual nozzle outside diameter.		

4.5.20 Figures



Determine the contributing areas as applicable where L_R, L_H, and L_I are determined from the procedures in paragraphs 4.5.6 and 4.5.11.

= A_1 = Area contributed by shell

= A_2 = Area contributed by nozzle projecting outward

= A_3 = Area contributed by nozzle projecting inward

= A_{41} = Area contributed by outward weld

= A_{42} = Area contributed by pad to vessel weld

= A_{43} = Area contributed by inward weld

= A_5 = Area contributed by reinforcing pad

A_T = Total area contributed

Notes: 1. Do not include any area that falls outside of the limits defined by L_H, L_R, and L_I. For example, if $W \geq L_R$, then $W = L_R$ and $A_{42} = 0.0$.
2. In accordance paragraph 4.1.4.1, all dimensions are in the corroded condition.

Figure 4.5.1 – Nomenclature for Reinforced Openings

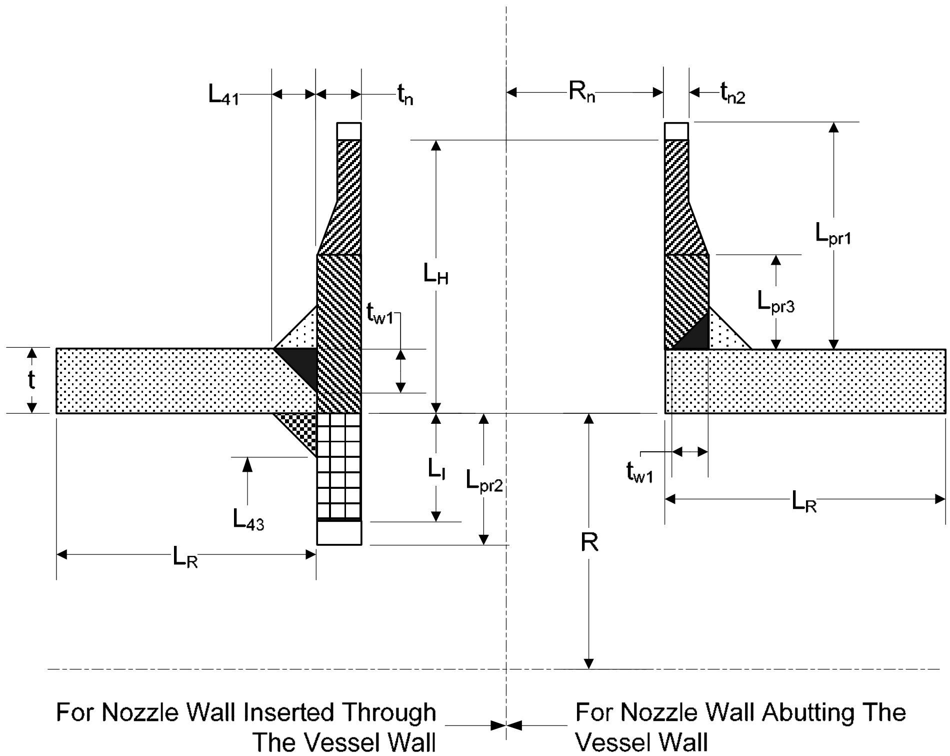


Figure 4.5.2 – Nomenclature for Variable Thickness Openings

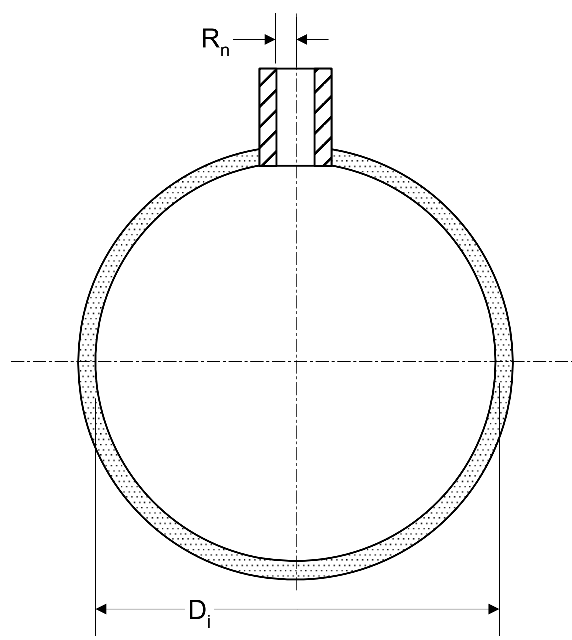


Figure 4.5.3 – Radial Nozzle in a Cylindrical Shell

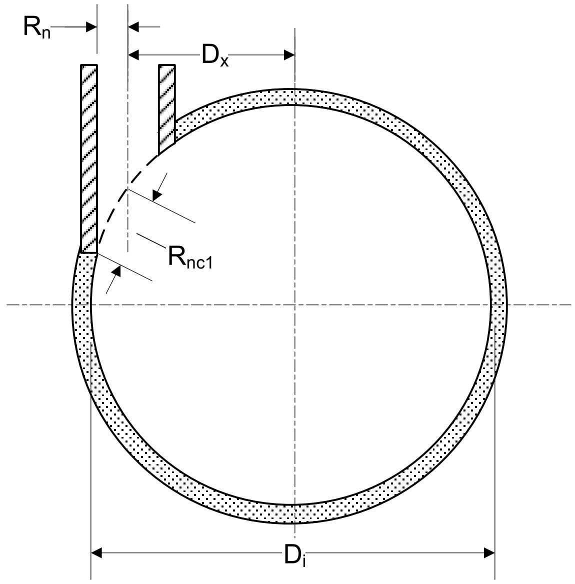


Figure 4.5.4 – Hillside Nozzle in a Cylindrical Shell

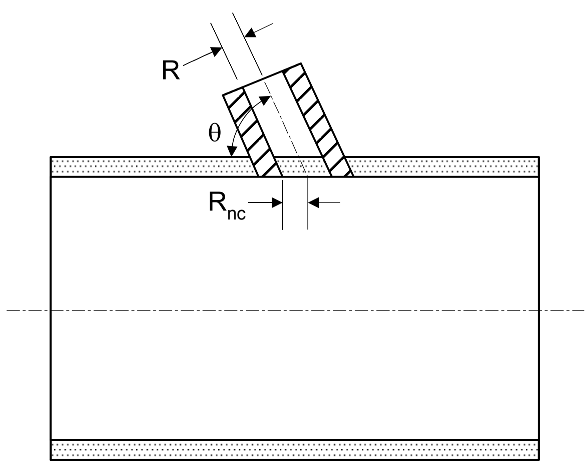


Figure 4.5.5 – Nozzle in a Cylindrical Shell Oriented at an Angle from the Longitudinal Axis

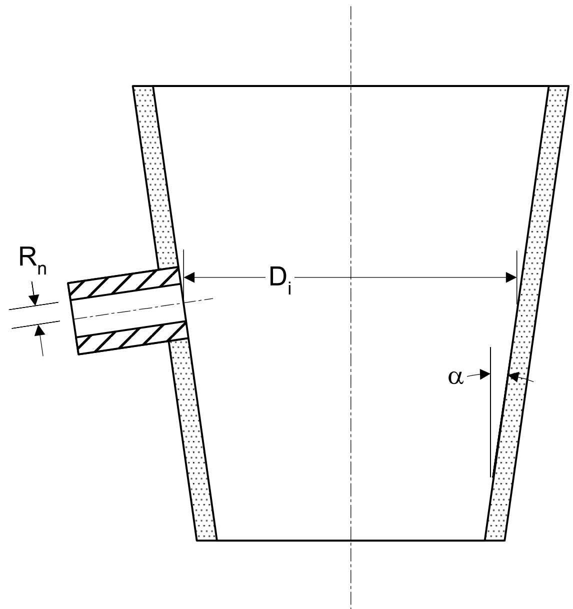


Figure 4.5.6 – Radial Nozzle in a Conical Shell

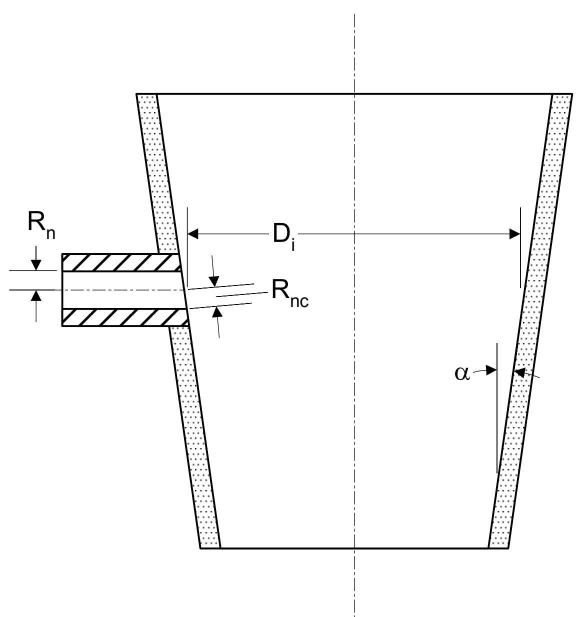


Figure 4.5.7 – Nozzle in a Conical Shell Oriented Perpendicular to the Longitudinal Axis

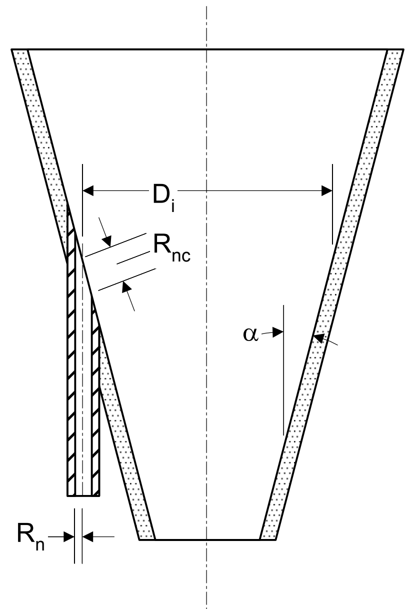


Figure 4.5.8 – Nozzle in a Conical Shell Oriented Parallel to the Longitudinal Axis

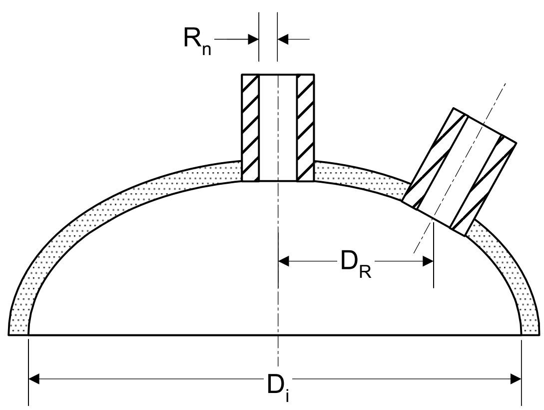


Figure 4.5.9 – Radial Nozzle in a Formed Head

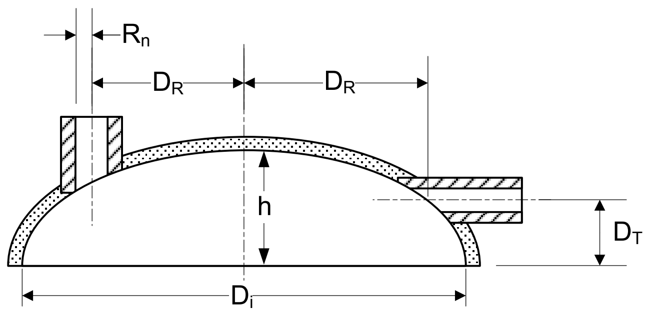


Figure 4.5.10 – Hillside or Perpendicular Nozzle in a Formed Head

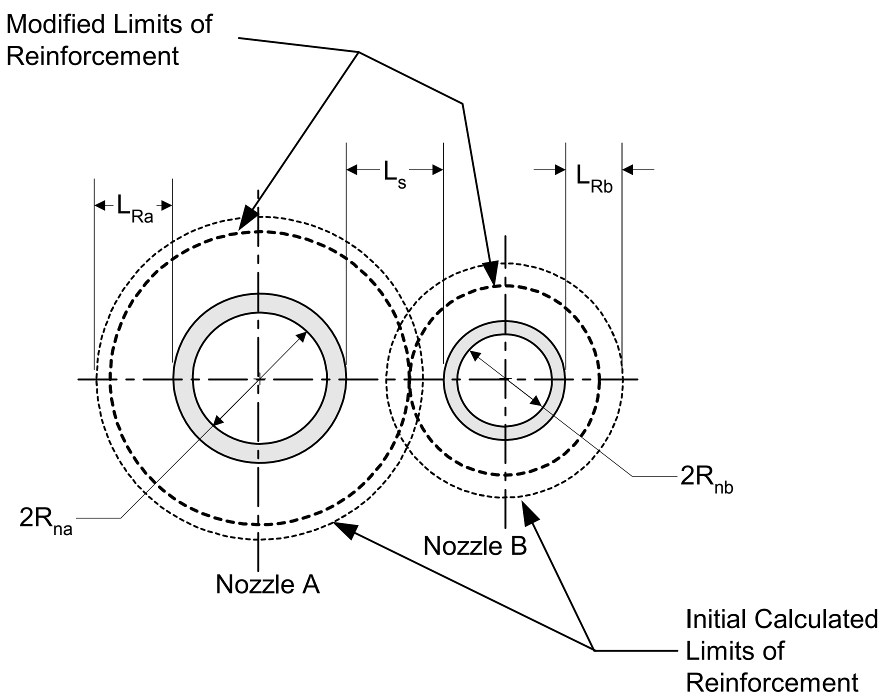


Figure 4.5.11 – Example of Two Adjacent Nozzle Openings

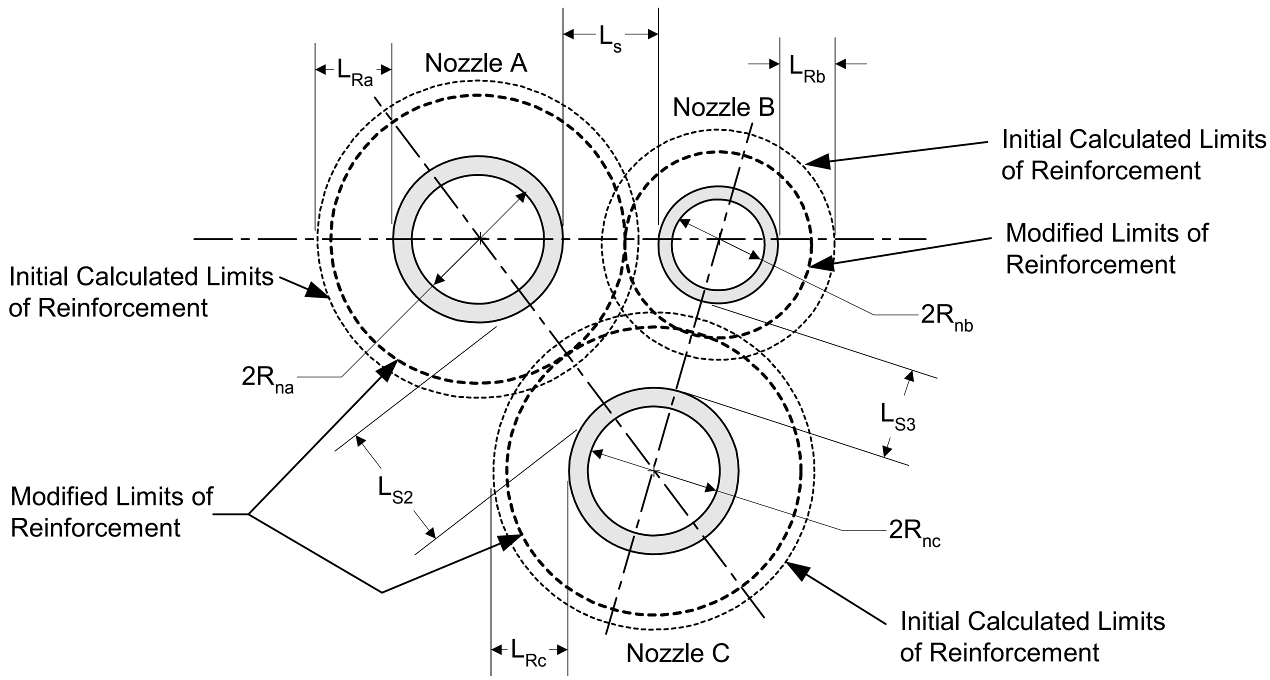


Figure 4.5.12 – Example of Three Adjacent Nozzle Openings

4.6 Design Rules for Flat Heads

4.6.1 Scope

4.6.1.1 The minimum thickness of unstayed flat heads, cover plates and blind flanges shall conform to the requirements given in paragraph 4.6. These requirements apply to both circular and noncircular heads and covers. Some acceptable types of flat heads and covers are shown in Table 4.6.1. In this table, the dimensions of the component parts and the dimensions of the welds are exclusive of extra metal required for corrosion allowance.

4.6.1.2 The design methods in this paragraph provide adequate strength for the design pressure. A greater thickness may be necessary if a deflection criterion is required for operation (e.g. leakage at threaded or gasketed joints).

4.6.1.3 For flat head types with a bolted flange connection where the gasket is located inside the bolt circle, calculations shall be made for two design conditions, gasket seating and operating conditions. Details regarding computation of design bolt loads for these two conditions are provided in paragraph 4.16.

4.6.2 Flat Unstayed Circular Heads

4.6.2.1 Circular blind flanges conforming to any of the flange standards listed in Part 1 and the requirements of paragraph 4.1.11 are acceptable for the diameters and pressure-temperature ratings in the respective standard when the blind flange is of the types shown in Table 4.6.1, Detail 7.

4.6.2.2 The minimum required thickness of a flat unstayed circular head or cover that is not attached with bolting that results in an edge moment shall be calculated by the following equation.

$$t = d\sqrt{\frac{CP}{S_{ho}E}} \tag{4.6.1}$$

4.6.2.3 The minimum required thickness of a flat unstayed circular head, cover, or blind flange that is attached with bolting that results in an edge moment (see Table 4.6.1, Detail 7) shall be calculated by the equations shown below. The operating and gasket seating bolt loads, W_o and W_g, and the moment arm of this load, h_G, in these equations shall be computed based on the flange geometry and gasket material as described in paragraph 4.16.

$$t = \max\left[t_o,\, t_g\right] \tag{4.6.2}$$

where

$$t_o = d\sqrt{\frac{CP}{S_{ho}E} + \frac{1.9W_o h_G}{S_{ho}Ed^3}} \tag{4.6.3}$$

$$t_g = d\sqrt{\frac{1.9W_g h_G}{S_{hg}Ed^3}} \tag{4.6.4}$$

4.6.3 Flat Unstayed Non-Circular Heads

4.6.3.1 The minimum required thickness of a flat unstayed non-circular head or cover that is not attached with bolting that results in an edge moment shall be calculated by the following equation.

$$t = d\sqrt{\frac{ZCP}{S_{ho}E}} \tag{4.6.5}$$

where

$$Z = \min\left[2.5, \left(3.4 - \frac{2.4d}{D}\right)\right] \tag{4.6.6}$$

4.6.3.2 The minimum required thickness of a flat unstayed non-circular head, cover, or blind flange that is attached with bolting that results in an edge moment (see Table 4.6.1, Detail 7) shall be calculated by the equations shown below. The operating and gasket seating bolt loads, W_o and W_g, and the moment arm of this load, h_G, in these equations shall be computed based on the flange geometry and gasket material as described in paragraph 4.16.

$$t = \max\left[t_o,\, t_g\right] \tag{4.6.7}$$

where

$$t_o = d\sqrt{\frac{ZCP}{S_{ho}E} + \frac{6W_o h_G}{S_{ho}ELd^2}} \tag{4.6.8}$$

$$t_g = d\sqrt{\frac{6W_g h_G}{S_{hg}ELd^2}} \tag{4.6.9}$$

The parameter Z is given by Equation (4.6.6).

4.6.4 Integral Flat Head With A Centrally Located Opening

4.6.4.1 Flat heads which have a single, circular, centrally located opening that exceeds one-half of the head diameter shall be designed in accordance with the rules which follow. A general arrangement of an integral flat head with or without a nozzle attached at the central opening is shown in Figure 4.6.1.

a) The shell-to-flat head juncture shall be integral, as shown in Table 4.6.1, Details 1, 2, 3, and 4. Alternatively, a butt weld, or a full penetration corner weld similar to the joints shown in Table 4.6.1 Details 5 and 6 may be used.

b) The central opening in the flat head may have a nozzle that is integral or integrally attached by a full penetration weld, or a nozzle attached by non-integral welds (i.e.: a double fillet or partial penetration weld, or may have an opening without an attached nozzle or hub. In the case of a nozzle attached by non-integral welds, the head is designed as a head without an attached nozzle or hub.

4.6.4.2 The head thickness does not have to meet the rules in paragraphs 4.6.2 or 4.6.3. The flat head thickness and other geometry parameters need only satisfy the allowable stress limits in Table 4.6.3.

4.6.4.3 A procedure that can be used to design an integral flat head with a single, circular centrally located opening is shown below.

a) STEP 1 – Determine the design pressure and temperature of the flat head opening.

b) STEP 2 – Determine the geometry of the flat head opening (see Figure 4.6.1).

c) STEP 3 – Calculate the operating moment, M_o, using the following equation.

$$M_o = 0.785 B_n^2 P\left(R + \frac{g_{1n}}{2}\right) + 0.785(B_s^2 - B_n^2)P\left(\frac{R + g_{1n}}{2}\right) \tag{4.6.10}$$

where

$$R = \frac{B_s - B_n}{2} - g_{1n} \tag{4.6.11}$$

d) STEP 4 – Calculate F, V, and f based on B_n, g_{1n}, g_{0n} and h_n using the equations in Table 4.16.4 and Table 4.16.5, designate the resulting values as F_n, V_n, and f_n.

e) STEP 5 – Calculate F, V, and f based on B_s, g_{1s}, g_{0s} and h_s using the equations in Table 4.16.4 and Table 4.16.5, designate the resulting values as F_s, V_s, and f_s.

f) STEP 6 – Calculate Y, T, U, Z, L, e, and d based on $K = A/B_n$ using the equations in Table 4.16.4.

g) STEP 7 – Calculate the quantity $(E\theta)^*$ using one of the following equations:

For an opening with an integrally attached nozzle:

$$(E\theta)^* = \frac{0.91\left(\frac{g_{1n}}{g_{on}}\right)^2 (B_n + g_{0n})V_n}{f_n\sqrt{B_n g_{0n}}} S_H \tag{4.6.12}$$

Where S_H is evaluated using the equation in Table 4.6.2.

For an opening without an attached nozzle or with a nozzle or hub attached with non-integral welds:

$$(E\theta)^* = \frac{B_n S_T}{t} \tag{4.6.13}$$

Where S_T is evaluated using the equation in Table 4.6.2.

h) STEP 8 – Calculate the quantity M_H using the following equation:

$$M_H = \frac{(E\theta)^*}{\frac{1.74 V_s\sqrt{B_s g_{0s}}}{g_{0s}^3(B_s + g_{0s})} + \frac{(E\theta)^*}{M_o}\left(1 + \frac{F_s t}{\sqrt{B_s g_{0s}}}\right)} \tag{4.6.14}$$

i) STEP 9 – Calculate the quantity X_1 using the following equation:

$$X_1 = \frac{M_o - M_H\left(1 + \frac{F_s t}{\sqrt{B_s g_{0s}}}\right)}{M_o} \tag{4.6.15}$$

j) STEP 10 – Calculate the stresses at the shell-to-flat head junction and opening-to-flat-head junction using Table 4.6.2.

k) STEP 11 – Check the flange stress acceptance criteria in Table 4.6.3. If the stress criteria are satisfied, then the design is complete. If the stress criteria are not satisfied, then re-proportion the flat head and/or opening dimensions and go to STEP 3.

4.6.5 Nomenclature

A shell outside diameter.

B_s inside diameter of the shell.

B_n inside diameter of the opening.

C factor depending upon the method of attachment of head, shell dimensions, and other items as described in Table 4.6.1. It should be noted that the value of C for a welded cover includes a factor of 0.667 that effectively increases the allowable stress for such constructions to 1.5S.

D is the long span of noncircular heads or covers measured perpendicular to short span.

d diameter, or short span, measured as indicated in figure shown in Table 4.6.1.

E joint factor.

e flange stress factor.

f_n hub stress correction factor for the nozzle opening-to-flat head junction.

f_s hub stress correction factor the shell-to-flat head junction

E weld joint factor (see paragraph 4.2).

F_n flange stress factor for the nozzle opening-to-flat head junction.

F_s flange stress factor for the shell-to-flat head junction

g_{1s} hub thickness at the large end of the shell-to-flat head junction.

g_{0n} hub thickness at the small end of the nozzle opening-to-flat head junction.

g_{0s} hub thickness at the small end of the shell-to-flat head junction.

g_{1n} hub thickness at the large end of the nozzle opening-to-flat head junction.

g_{1s} hub thickness at the large end of the shell-to-flat head junction.

h_G gasket moment arm (see Table 4.16.6).

h_n hub length at the large end of the nozzle opening-to-flat head junction.

h_s hub length at the large end of the shell-to-flat head junction.

L perimeter of a noncircular bolted head measured along the centers of the bolt holes, or the flange stress factor, as applicable.

M_o operating moment.

M_H moment acting at the shell-to-flat head junction.

m thickness ratio t_r/t_s .

P internal design pressure.

r inside corner radius on a head formed by flanging or forging.

S_{ho} allowable stress from Annex 3.A for the head evaluated at the design temperature.

S_{hg} allowable stress from Annex 3.A for the head evaluated at the gasket seating condition.

T flange stress factor.

t minimum required thickness of the flat head or cover.

t_g required thickness of the flat head or cover for the gasket seating condition.

t_o required thickness of the flat head or cover for the design operating condition.

t_f nominal thickness of the flange on a forged head at the large end.

t_h nominal thickness of the flat head or cover.

t_r required thickness of a seamless shell.

t_s nominal thickness of the shell.

t_1 throat dimension of the closure weld

U flange stress factor.

V_n flange stress factor for the nozzle opening-to-flat head junction.

V_s flange stress factor for the shell-to-flat head junction

W_o operating bolt load at the design operating condition.

W_g gasket seating bolt load at the design gasket seating condition.

Y length of the flange of a flanged head, measured from the tangent line of knuckle, or the flange stress factor, as applicable.

Z factor for noncircular heads and covers that depends on the ratio of short span to long span, or the flange stress factor, as applicable.

Z_1 integral flat head stress parameter.

$(E\theta)^*$ slope of head with central opening or nozzle times the modulus of elasticity, disregarding the interaction of the integral shell at the outside diameter of the head.

4.6.6 Tables

Table 4.6.1 – C Parameter for Flat Head Designs

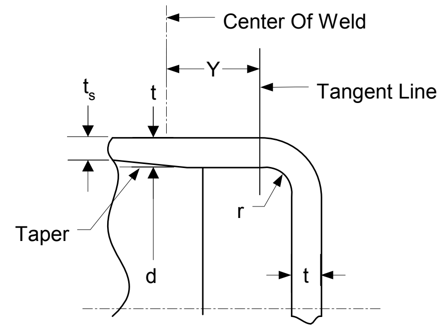
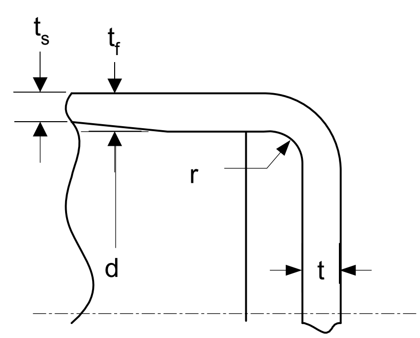
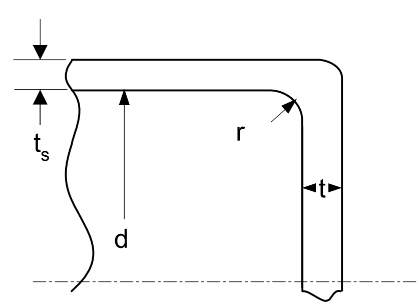
Detail	Requirements	Figure
1	• $C = 0.17$ for flanged circular and noncircular heads forged integral with or butt welded to the vessel with an inside corner radius not less than three times the required head thickness, with no special requirement with regard to length of flange. • $C = 0.10$ for circular heads, when the flange length for heads of the above design is not less than: $Y = \left(1.1 - 0.8\left(\frac{t_s}{t_h}\right)^2\right)\sqrt{dt_h}$ • $C = 0.10$ for circular heads, when the flange length Y less than the requirements in the above equation but the shell thickness is not less than: $t_s = 1.12t_h\sqrt{1.1 - Y/\sqrt{dt_h}}$ for a length of at least $2\sqrt{dt_s}$. When $C = 0.10$ is used, the taper shall be at least 1:3. • $r = 3t$ minimum shall be used	
2	• $C = 0.17$ for forged circular and noncircular heads integral with or butt welded to the vessel, where the flange thickness is not less than two times the shell thickness, the corner radius on the inside is not less than three times the flange thickness. • $r = 3t_f$ minimum shall be used	
3	• $C = \max\left[0.33m,\ 0.20\right]$ for forged circular and noncircular heads integral with or butt welded to the vessel, where the flange thickness is not less than the shell thickness, the corner radius on the inside is not less than the following: $r = 10\ mm\ (0.375\ in.)$ $for\ t_s \le 38\ mm\ (1.5\ in.)$ $r = \min\left[0.25t_s,\ 19\ mm\ \left(0.75\ in.\right)\right]$ $for\ t_s > 38\ mm\ (1.5\ in.)$	

Table 4.6.1 – C Parameter for Flat Head Designs

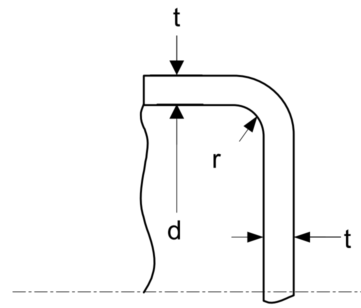
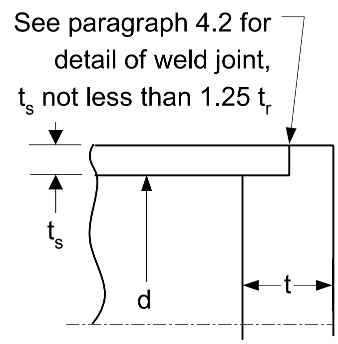
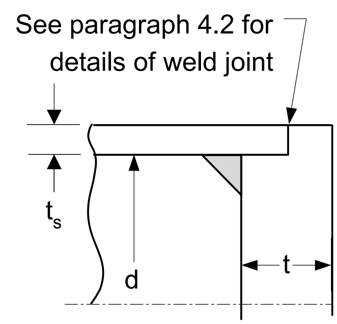
Detail	Requirements	Figure
4	• $C = 0.13$ for integral flat circular heads when: – the dimension d does not exceed 610 mm (24 in.) – the ratio of thickness of the head to the dimension d is not less than 0.05 or greater than 0.25 – the head thickness t_h is not less than the shell thickness t_s – the inside corner radius is not less than $0.25t$ – the construction is obtained by special techniques of upsetting and spinning the end of the shell, such as employed in closing header ends. • $r = 3t$ minimum shall be used	
5	$C = 0.33$ for circular plates welded to the end of the shell when t_s is at least $1.25t_r$ and the weld details conform to the requirements of paragraph 4.2.	
6	$C = \max\left[0.33m,\ 0.20\right]$ for circular plates if an inside fillet weld with minimum throat thickness of $0.7t_s$ is used and the details of the outside weld conform to the requirements of paragraph 4.2.	

Table 4.6.1 – C Parameter for Flat Head Designs

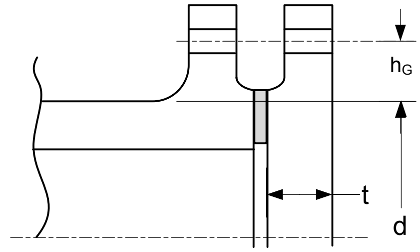
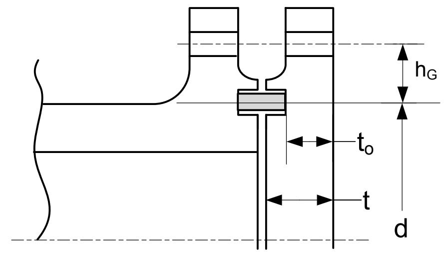
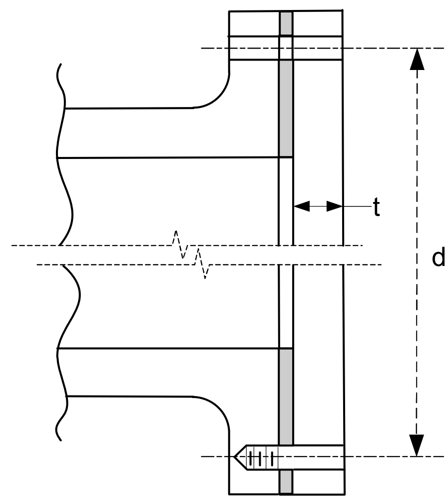
Detail	Requirements	Figure
7	• $C = 0.3$ for circular and noncircular heads and covers bolted to the vessel as indicated in the figures. • When the cover plate is grooved for a peripheral gasket, the net cover plate thickness under the groove or between the groove and the outer edge of the cover plate shall be not less than the following thickness. For circular heads and covers: $t_o = d\sqrt{\frac{1.9W_o h_G}{S_{ho} d^3}}$ For noncircular heads and covers: $t_o = d\sqrt{\frac{6W_o h_G}{S_{ho} L d^2}}$	 
8	$C = 0.25$ for circular covers bolted with a full-face gasket to shells and flanges.	

Table 4.6.1 – C Parameter for Flat Head Designs

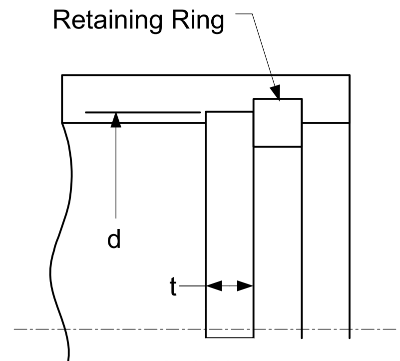
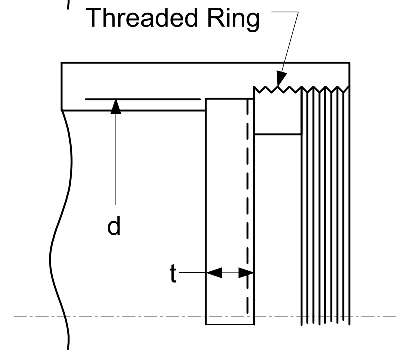
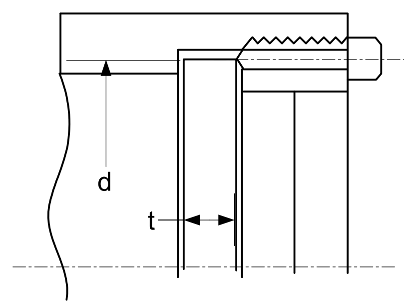
Detail	Requirements	Figure
9	$C = 0.3$ for a circular plate inserted into the end of a vessel and held in place by a positive mechanical locking arrangement when all possible means of failure (either by shear, tension, compression, or radial deformation, including flaring, resulting from pressure and differential thermal expansion) are resisted with a design factor of at least four. Seal welding may be used, if desired.	  

Table 4.6.2 – Junction Stress Equations for an Integral Flat Head With Opening

Head/Shell Junction Stresses	Opening/Head Junction Stresses
$S_{HS} = \dfrac{1.1 f_s X_1 \left(E\theta\right)^* \sqrt{B_s g_{0s}}}{\left(\dfrac{g_{1s}}{g_{0s}}\right)^2 B_s V_s}$ $S_{RS} = \dfrac{1.91 M_H \left(1 + \dfrac{F_s t}{\sqrt{B_s g_{0s}}}\right)}{B_s t^2} + \dfrac{0.64 F_s M_H}{B_s \sqrt{B_s g_{0s}}\, t}$ $S_{TS} = \dfrac{X_1 \left(E\theta\right)^* t}{B_s} - \dfrac{0.57 M_H \left(1 + \dfrac{F_s t}{\sqrt{B_s g_{0s}}}\right)}{B_s t^2} + \dfrac{0.64 Z F_s M_H}{B_s \sqrt{B_s g_{0s}}\, t}$	$S_{HO} = X_1 S_H$ $S_{RO} = X_1 S_R$ $S_{TO} = X_1 S_T + \dfrac{0.64 Z_1 F_s M_H}{B_s \sqrt{B_s g_{0s}}\, t}$ where $S_H = \dfrac{f_n M_o}{L g_{1n}^2 B_n}$ $S_R = \dfrac{\left(1.33 te + 1\right) M_o}{L t^2 B_n}$ $S_T = \dfrac{Y M_o}{t^2 B_n} - Z S_R$ $Z_1 = \dfrac{2K^2}{K^2 - 1}$ Note: $S_R = S_H = 0.0$ for the case of an opening without a nozzle

Table 4.6.3 – Stress Acceptance Criteria for an Integral Flat Head With Opening

Head/Shell Junction Stresses	Opening/Head Junction Stresses
$S_{HS} \le 1.5 S_{ho}$	$S_{HO} \le 1.5 S_{ho}$
$S_{RS} \le S_{ho}$	$S_{RO} \le S_{ho}$
$S_{TS} \le S_{ho}$	$S_{TO} \le S_{ho}$
$\frac{\left(S_{HS} + S_{RS}\right)}{2} \le S_{ho}$	$\frac{\left(S_{HO} + S_{RO}\right)}{2} \le S_{ho}$
$\frac{\left(S_{HS} + S_{TS}\right)}{2} \le S_{ho}$	$\frac{\left(S_{HO} + S_{TO}\right)}{2} \le S_{ho}$

4.6.7 Figures

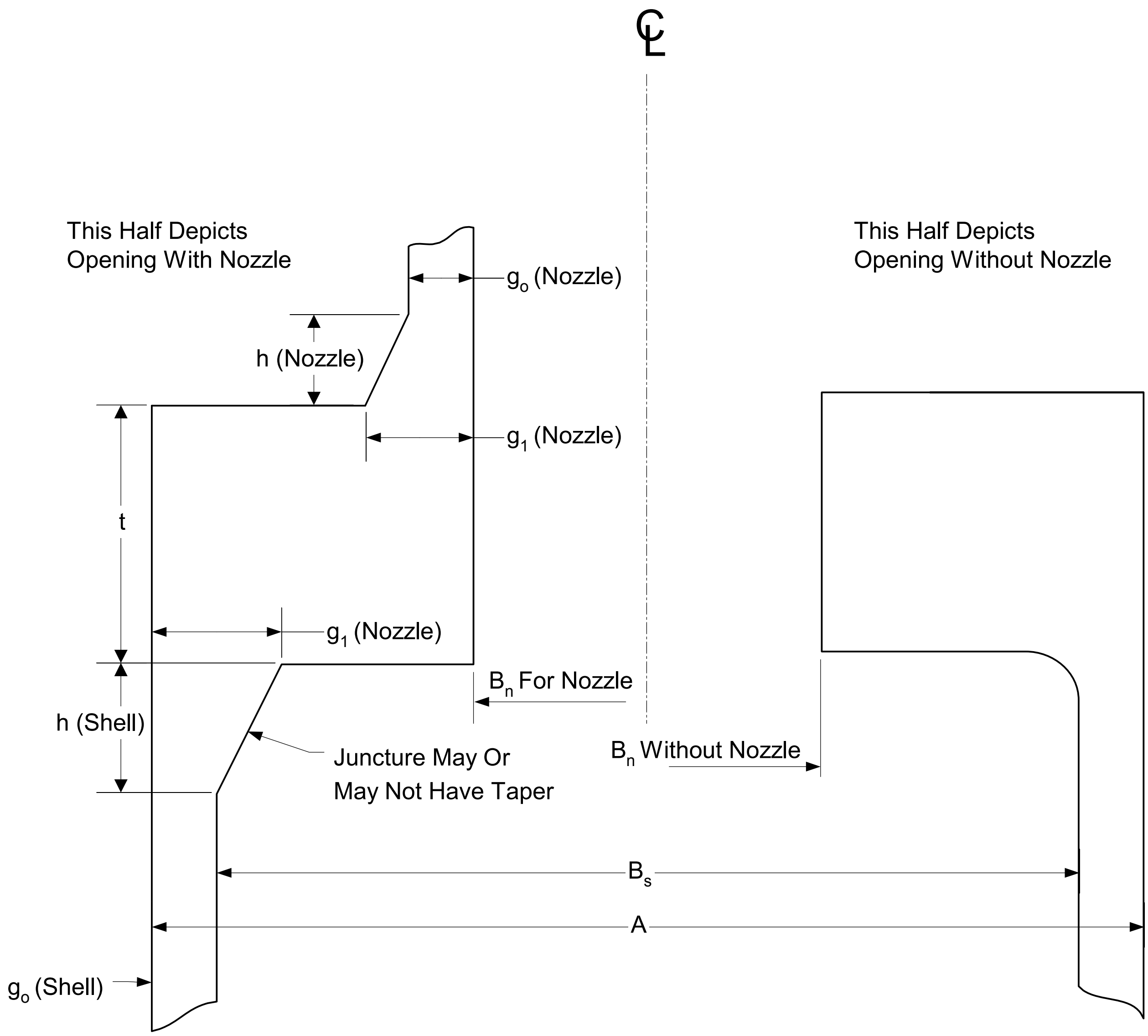


Figure 4.6.1 – Integral Flat head With a Large Central Opening

4.7 Design Rules for Spherically Dished Bolted Covers

4.7.1 Scope

4.7.1.1 Design rules for four configurations of circular spherically dished heads with bolting flanges are provided in paragraph 4.7. The four head types are shown in Figures 4.7.1, 4.7.2, 4.7.3, and 4.7.4. The design rules cover both internal and external pressure, pressure that is concave and convex to the spherical head, respectively. The maximum value of the pressure differential shall be used in all of the equations.

4.7.1.2 For head types with a bolted flange connection where the gasket is located inside the bolt circle, calculations shall be made for two design conditions, gasket seating and operating conditions. Details regarding computation of design bolt loads and flange moments for these two conditions are provided in paragraph 4.16. If a flange moment is computed as a negative number, the absolute value of this moment shall be used in all of the equations.

4.7.1.3 Calculations shall be performed using dimensions in the corroded condition and the uncorroded condition, and the more severe case shall control.

4.7.2 Type A Head Thickness Requirements

4.7.2.1 The thickness of the head and skirt for a Type A Head Configuration (see Figure 4.7.1) shall be determined in accordance with the rules in paragraph 4.3 for internal pressure (pressure on the concave side), and paragraph 4.4 for external pressure (for pressure on the convex side). The skirt thickness shall be determined using the appropriate formula for cylindrical shells. The head radius, L, and knuckle radius, r, shall comply with the limitations given in these paragraphs.

4.7.2.2 The flange thickness of the head for a Type A Head Configuration shall be determined in accordance with the rules of paragraph 4.16. When a slip-on flange conforming to the standards listed in Table 1.1 is used, design calculations per paragraph 4.16 need not be done provided the design pressure-temperature is within the pressure-temperature rating permitted in the flange standard.

4.7.2.3 Detail (a) in Figure 4.7.1 is permitted if both of the following requirements are satisfied.

a) The material of construction satisfies the following equation.

$$\frac{S_{yT}}{S_u} \le 0.625 \tag{4.7.1}$$

b) The component is not in cyclic service, i.e. a fatigue analysis is not required in accordance with paragraph 4.1.1.4.

4.7.3 Type B Head Thickness Requirements

4.7.3.1 The thickness of the head for a Type B Head Configuration (see Figure 4.7.2) shall be determined by the following equations.

a) Internal pressure (pressure on the concave side)

$$t = \frac{5PL}{6S} \tag{4.7.2}$$

b) External pressure (pressure on the convex side) – the head thickness shall be determined in accordance with the rules in paragraph 4.4.

4.7.3.2 The flange thickness of the head for a Type B Head Configuration shall be determined by the following equations where the flange moments M_o and M_g for the operating and gasket seating conditions, respectively, are determined from paragraph 4.16.

a) Flange thickness for a ring gasket

$$T = \max\left[T_g, T_o\right] \tag{4.7.3}$$

where

$$T_g = \sqrt{\frac{M_g}{S_{fg}B}\left(\frac{A+B}{A-B}\right)} \tag{4.7.4}$$

$$T_o = \sqrt{\frac{M_o}{S_{fo}B}\left(\frac{A+B}{A-B}\right)} \tag{4.7.5}$$

b) Flange thickness for a full face gasket

$$T = 0.6\sqrt{\frac{|P|}{S_{fo}}\left(\frac{B(A+B)(C-B)}{A-B}\right)} \tag{4.7.6}$$

4.7.3.3 A Type B head may only be used if both of the requirements in paragraph 4.7.2.3 are satisfied.

4.7.4 Type C Head Thickness Requirements

4.7.4.1 The thickness of the head for a Type C Head Configuration (see Figure 4.7.3) shall be determined by the following equations.

a) Internal pressure (pressure on the concave side) – the head thickness shall be determined using Equation (4.7.2).

b) External pressure (pressure on the convex side) – the head thickness shall be determined in accordance with the rules in paragraph 4.4.

4.7.4.2 The flange thickness of the head for a Type C Head Configuration shall be determined by the following equations where the flange moments M_o and M_g for the operating and gasket seating conditions, respectively, are determined from paragraph 4.16.

a) Flange thickness for a ring gasket for heads with round bolting holes

$$T = \max\left[T_g, T_o\right] \tag{4.7.7}$$

where

$$T_g = \sqrt{\frac{1.875M_g}{S_{fg}B}\left(\frac{C+B}{7C-5B}\right)} \tag{4.7.8}$$

$$T_o = Q + \sqrt{\frac{1.875M_o}{S_{fo}B}\left(\frac{C+B}{7C-5B}\right)} \tag{4.7.9}$$

$$Q = \frac{|P|L}{4S_{fo}}\left(\frac{C+B}{7C-5B}\right) \tag{4.7.10}$$

b) Flange thickness for ring gasket for heads with bolting holes slotted through the edge of the head

$$T = \max\left[T_g, T_o\right] \tag{4.7.11}$$

where

$$T_g = \sqrt{\frac{1.875M_g}{S_{fg}B}\left(\frac{C+B}{3C-B}\right)} \tag{4.7.12}$$

$$T_o = Q + \sqrt{\frac{1.875M_o}{S_{fo}B}\left(\frac{C+B}{3C-B}\right)} \tag{4.7.13}$$

$$Q = \frac{|P|L}{4S_{fo}}\left(\frac{C+B}{3C-B}\right) \tag{4.7.14}$$

c) Flange thickness for full-face gasket for heads with round bolting holes

$$T = Q + \sqrt{Q^2 + \frac{3BQ(C-B)}{L}} \tag{4.7.15}$$

The parameter Q is given by Equation (4.7.10).

d) Flange thickness for full-face gasket for heads with bolting holes slotted through the edge of the head

$$T = Q + \sqrt{Q^2 + \frac{3BQ(C-B)}{L}} \tag{4.7.16}$$

The parameter Q is given by Equation (4.7.14).

4.7.5 Type D Head Thickness Requirements

4.7.5.1 The thickness of the head for a Type D Head Configuration (see Figure 4.7.4) shall be determined by the following equations.

a) Internal pressure (pressure on the concave side) – the head thickness shall be determined using Equation (4.7.2).

b) External pressure (pressure on the convex side) – the head thickness shall be determined in accordance with the rules in paragraph 4.4.

4.7.5.2 The flange thickness of the head for a Type D Head Configuration shall be determined by the following equations.

$$T = \max\left[T_g, T_o\right] \tag{4.7.17}$$

where

$$T_g = \sqrt{\frac{M_g}{S_{fg}B}\left(\frac{A+B}{A-B}\right)} \tag{4.7.18}$$

$$T_o = Q + \sqrt{Q^2 + \frac{M_o}{S_{fo}B}\left(\frac{A+B}{A-B}\right)} \tag{4.7.19}$$

$$Q = \frac{|P| B\sqrt{4L^2 - B^2}}{8S_{fo}\left(A-B\right)} \tag{4.7.20}$$

When determining the flange design moment for the design condition, M_o, using paragraph 4.16, the following modifications shall be made. The moment arm, h_D, shall be computed using Equation (4.7.21). An additional moment term, M_r, computed using Equation (4.7.22) shall be added to M_o as defined paragraph 4.16. The term M_{oe} in the equation for M_o as defined paragraph 4.16 shall be set to zero in this calculation. Note that this term may be positive or negative depending on the orientation of $t_v,\ R,\ A_R$.

$$h_D = 0.5\left(C - B\right) \tag{4.7.21}$$

$$M_r = \left(0.785B^2 P\cot\left[\beta\right]\right)h_r \tag{4.7.22}$$

where

$$\beta = \arcsin\left[\frac{B}{2L+t}\right] \tag{4.7.23}$$

4.7.5.3 As an alternative to the rules in paragraphs 4.7.5.1 and 4.7.5.2, the following procedure can be used to determine the required head and flange thickness of a Type D head. This procedure accounts for the continuity between the flange ring and the head, and represents a more accurate method of analysis.

a) STEP 1 – Determine the design pressure and temperature of the flange joint. If the pressure is negative, a negative value must be used for P in all of the equations of this procedure, and

$$P_e = 0.0 \qquad \textit{for internal pressure} \tag{4.7.24}$$

$$P_e = P \qquad \textit{for external pressure} \tag{4.7.25}$$

b) STEP 2 – Determine an initial Type D head configuration geometry (see Figure 4.7.5). The following geometry parameters are required:

1) The flange bore, B
2) The bolt circle diameter, C
3) The outside diameter of the flange, A
4) Flange thickness, T
5) Mean head radius, R
6) Head thickness, t

7) Inside depth of flange to the base of the head, q

c) STEP 3 – Select a gasket configuration and determine the location of the gasket reaction, G, and the design bolt loads for the gasket seating, W_g, and operating conditions, W_o, using the rules of paragraph 4.16.

d) STEP 4 – Determine the geometry parameters.

$$h_1 = \frac{(C-G)}{2} \tag{4.7.26}$$

$$h_2 = \frac{(G-B)}{2} \tag{4.7.27}$$

$$d = \frac{(A-B)}{2} \tag{4.7.28}$$

$$n = \frac{T}{t} \tag{4.7.29}$$

$$K = \frac{A}{B} \tag{4.7.30}$$

$$\phi = \arcsin\left[\frac{B}{2R}\right] \tag{4.7.31}$$

$$e = q - \frac{1}{2}\left(T - \frac{t}{\cot[\phi]}\right) \tag{4.7.32}$$

$$k_1 = 1 - \left(\frac{1-2\nu}{2\lambda}\right)\cot[\phi] \tag{4.7.33}$$

$$k_2 = 1 - \left(\frac{1+2\nu}{2\lambda}\right)\cot[\phi] \tag{4.7.34}$$

$$\lambda = \left[3\left(1-\nu^2\right)\left(\frac{R}{t}\right)^2\right]^{0.25} \tag{4.7.35}$$

e) STEP 5 – Determine the shell discontinuity geometry factors.

$$C_1 = \frac{0.275n^3t \cdot \ln[K]}{k_1} - e \tag{4.7.36}$$

$$C_2 = \frac{1.1\lambda n^3 t \cdot \ln\left[K\right]}{Bk_1} + 1 \tag{4.7.37}$$

$$C_4 = \frac{\lambda \sin\left[\phi\right]}{2}\left(k_2 + \frac{1}{k_1}\right) + \frac{B}{4nd} + \frac{1.65e}{tk_1} \tag{4.7.38}$$

$$C_5 = \frac{1.65}{tk_1}\left(1 + \frac{4\lambda e}{B}\right) \tag{4.7.39}$$

f) STEP 6 – Determine the shell discontinuity load factors for the operating and gasket conditions.

$$C_{3o} = \frac{\pi B^2 P}{4}\left(e \cdot \cot\left[\phi\right] + \frac{2q\left(T-q\right)}{B} - h_2\right) - W_o h_1 \tag{4.7.40}$$

$$C_{6o} = \frac{\pi B^2 P}{4}\left(\frac{4q - B \cdot \cot\left[\phi\right]}{4nd} - \frac{0.35}{\sin\left[\phi\right]}\right) \tag{4.7.41}$$

$$C_{3g} = -W_g h_1 \tag{4.7.42}$$

$$C_{6g} = 0.0 \tag{4.7.43}$$

g) STEP 7 – Determine the shell discontinuity force and moment for the operating and gasket conditions.

$$V_{do} = \frac{C_2 C_{6o} - C_{3o} C_5}{C_2 C_4 - C_1 C_5} \tag{4.7.44}$$

$$M_{do} = \frac{C_1 C_{6o} - C_{3o} C_4}{C_2 C_4 - C_1 C_5} \tag{4.7.45}$$

$$V_{dg} = \frac{C_2 C_{6g} - C_{3g} C_5}{C_2 C_4 - C_1 C_5} \tag{4.7.46}$$

$$M_{dg} = \frac{C_1 C_{6g} - C_{3g} C_4}{C_2 C_4 - C_1 C_5} \tag{4.7.47}$$

h) STEP 8 – Calculate the stresses in the head and at the head-to-flange junction using Table 4.7.1 and check the stress acceptance criteria. If the stress criteria are satisfied, then the design is complete. If the stress criteria are not satisfied, then re-proportion the bolted head dimensions and go to STEP 3.

4.7.6 Nomenclature

A flange outside diameter.

B flange inside diameter.

β angle formed by the tangent to the center line of the dished cover thickness at its point of intersection with the flange ring, and a line perpendicular to the axis of the dished cover

C bolt circle diameter.

C_1 shell discontinuity geometry parameter for the Type D head alternative design procedure.

C_2 shell discontinuity geometry parameter for the Type D head alternative design procedure.

C_{3g} shell discontinuity load factor for the gasket seating condition for the Type D head alternative design procedure.

C_{3o} shell discontinuity load factor for the design operating condition for the Type D head alternative design procedure.

C_4 shell discontinuity geometry parameter for the Type D head alternative design procedure.

C_5 shell discontinuity geometry parameter for the Type D head alternative design procedure.

C_{6g} shell discontinuity load factor for the gasket seating condition for the Type D head alternative design procedure.

C_{6o} shell discontinuity load factor for the design operating condition for the Type D head alternative design procedure.

e geometry parameter for the Type D head alternative design procedure.

h_r moment arm of the head reaction force.

h_1 geometry parameter for the Type D head alternative design procedure.

h_2 geometry parameter for the Type D head alternative design procedure.

k_1 geometry parameter for the Type D head alternative design procedure.

k_2 geometry parameter for the Type D head alternative design procedure.

K geometry parameter for the Type D head alternative design procedure.

L inside crown radius.

λ geometry parameter for the Type D head alternative design procedure.

M_{dg} shell discontinuity moment for the gasket seating condition.

M_{do} shell discontinuity moment for design operating condition.

M_g flange design moment for the gasket seating condition determined using paragraph 4.16.

M_o flange design moment for the design condition determined using paragraph 4.16 (see paragraph 4.7.5.2 for exception)

M_r moment from the head reaction force.

n geometry parameter for the Type D head alternative design procedure.

ν is Poisson's ratio.

P design pressure.

P_e pressure factor to adjust the design rules for external pressure.

ϕ one-half central angle of the head for the Type D head alternative design procedure.

q inside depth of the flange to the base of the head.

R mean radius of a Type D head.

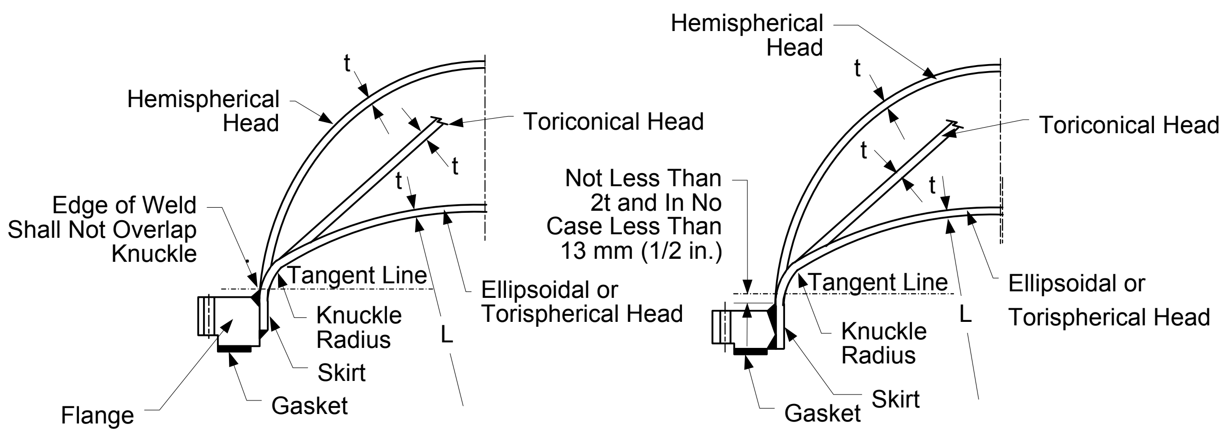
r	inside knuckle radius. S_{fg} allowable stress from Annex 3.A for the flange evaluated at the gasket seating condition.
S_{fm}	membrane stress in the flange.
S_{fmbi}	membrane plus bending stress on the inside surface of the flange.
S_{fmbo}	membrane plus bending stress on the outside surface of the flange.
S_{fo}	allowable stress from Annex 3.A for the flange evaluated at the design temperature.
S_{hb}	bending stress at the head-to-flange junction.
S_{hg}	allowable stress from Annex 3.A for the head evaluated at the gasket seating condition.
S_{hm}	head membrane stress.
S_{hl}	local membrane stress at the head-to-flange junction.
S_{hlbi}	local membrane plus bending stress at the head-to-flange junction on the inside surface of the head.
S_{hlbo}	local membrane plus bending stress at the head-to-flange junction on the outside surface of the head.
S_{ho}	allowable stress from Annex 3.A for the head evaluated at the design temperature.
S_{yT}	yield strength from Annex 3.D evaluated at the design temperature.
S_u	minimum specified ultimate tensile strength from Annex 3.D.
T	flange thickness.
$T*$	flange thickness for a Type C Head.
T_g	required flange thickness for the gasket seating condition.
T_o	required flange thickness for design operating condition.
t	required head thickness.
V_{dg}	shell discontinuity shear force for the gasket seating condition.
V_{do}	shell discontinuity shear force for design operating condition.
W_g	bolt load for the gasket seating condition.
W_o	bolt load for design operating condition.

4.7.7 Tables

Table 4.7.1 – Junction Stress Equations and Acceptance Criteria for a Type D Head

Operating Conditions	Gasket Seating Conditions
$S_{hm}=\frac{PR}{2t}+P_e$	$S_{hm}=0.0$
$S_{hl}=\frac{PR}{2t}+\frac{V_{do}\cos[\phi]}{\pi Bt}+P_e$	$S_{hl}=\frac{V_{dg}\cos[\phi]}{\pi Bt}$
$S_{hb}=\frac{6M_{do}}{\pi Bt^2}$	$S_{hb}=\frac{6M_{dg}}{\pi Bt^2}$
$S_{hlbi}=S_{hl}+S_{hb}$	$S_{hlbi}=S_{hl}+S_{hb}$
$S_{hlbo}=S_{hl}-S_{hb}$	$S_{hlbo}=S_{hl}-S_{hb}$
$S_{fm}=\frac{1}{\pi BT}\left(\frac{\pi B^2P}{4}\left(\frac{4q}{B}-\cot[\phi]\right)-V_{do}\right)\left(\frac{K^2+1}{K^2-1}\right)+P_e$	$S_{fm}=\frac{1}{\pi BT}\left(-V_{dg}\right)\left(\frac{K^2+1}{K^2-1}\right)$
$S_{fb}=\frac{0.525n}{Btk_1}\left(V_{do}-\frac{4M_{do}\lambda}{B}\right)$	$S_{fb}=\frac{0.525n}{Btk_1}\left(V_{dg}-\frac{4M_{dg}\lambda}{B}\right)$
$S_{fmbo}=S_{fm}-S_{fb}$	$S_{fmbo}=S_{fm}-S_{fb}$
$S_{fmbi}=S_{fm}+S_{fb}$	$S_{fmbi}=S_{fm}+S_{fb}$
Acceptance Criteria	
$S_{hm}\le S_{ho}$	$S_{hm}\le S_{hg}$
$S_{hl}\le 1.5S_{ho}$	$S_{hl}\le 1.5S_{hg}$
$S_{hlbi}\le 1.5S_{ho}$	$S_{hlbi}\le 1.5S_{hg}$
$S_{hlbo}\le 1.5S_{ho}$	$S_{hlbo}\le 1.5S_{hg}$
$S_{fm}\le S_{fo}$	$S_{fm}\le S_{fg}$
$S_{fmbo}\le 1.5S_{fo}$	$S_{fmbo}\le 1.5S_{fg}$
$S_{fmbi}\le 1.5S_{fo}$	$S_{fmbi}\le 1.5S_{fg}$

4.7.8 Figures



(a) Loose Flange Type (b) Integral Flange Type

Note: See Table 4.2.5, Details 2 and 3 for transition requirements for a head and skirt with different thicknesses

Figure 4.7.1 – Type A Dished Cover with a Bolting Flange

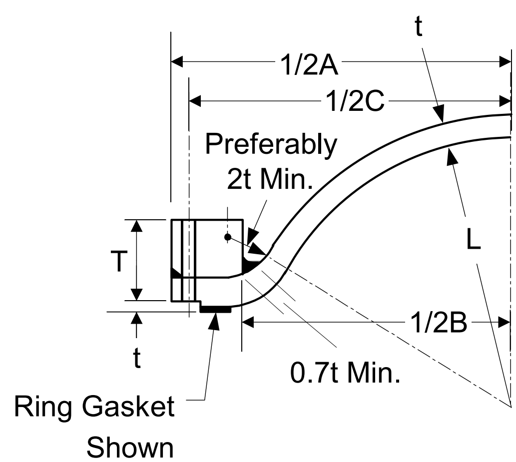


Figure 4.7.2 – Type B Spherically Dished Cover with a Bolting Flange

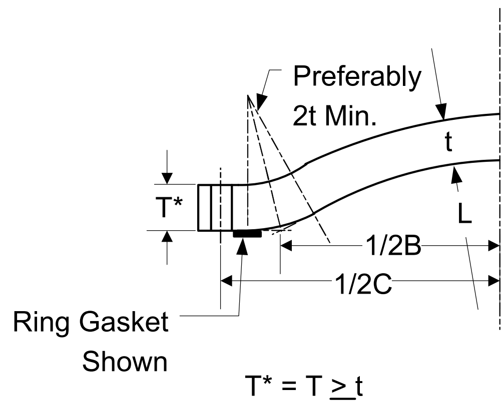


Figure 4.7.3 – Type C Spherically Dished Cover with a Bolting Flange

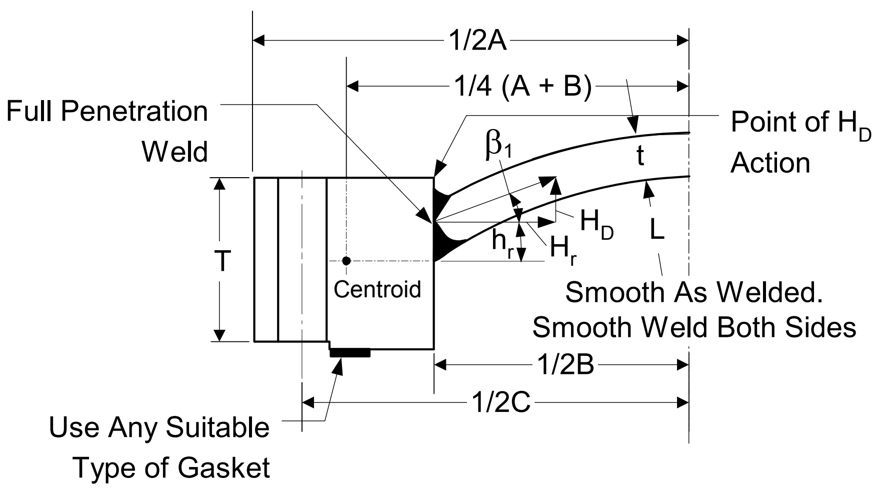


Figure 4.7.4 – Type D Spherically Dished Cover with a Bolting Flange

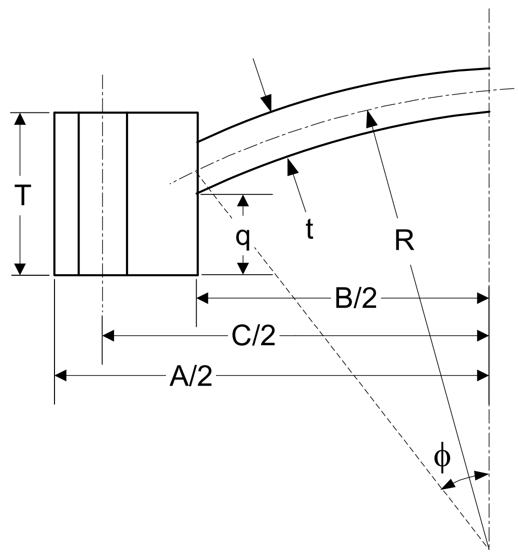


Figure 4.7.5 – Type D Head Geometry for Alternative Design Procedure

4.8 Design Rules for Quick-Actuating (Quick Opening) Closures

4.8.1 Scope

4.8.1.1 Design requirements for quick-actuating or quick-opening closures are provided in Paragraph 4.8. Specific calculation methods are not provided. However, the rules of Part 4 and Part 5 can be used to qualify the design of a quick-actuating or quick-opening closure.

4.8.2 Definitions

4.8.2.1 Quick-actuating or quick-opening closures are those that permit substantially faster access to the contents space of a pressure vessel than would be expected with a standard bolted flange connection (bolting through one or both flanges). Closures with swing bolts are not considered quick actuating (quick-opening).

4.8.2.2 Holding elements are structural members of the closure used to attach or hold the cover to the vessel, and/or to provide the load required to seal the closure. Hinge pins or bolts can be holding elements.

4.8.2.3 Locking components are parts of the closure that prevent a reduction in the load on a holding element that provides the force required to seal the closure, or prevent the release of a holding element. Locking components may also be used as holding elements.

4.8.2.4 The locking mechanism or locking device consists of a combination of locking components.

4.8.2.5 The use of a multi-link component, such as a chain, as a holding element is not permitted.

4.8.3 General Design Requirements

4.8.3.1 Quick-actuating closures shall be designed such that the locking elements will be engaged prior to or upon application of the pressure and will not disengage until the pressure is released.

4.8.3.2 Quick-actuating closures shall be designed such that the failure of a single locking component while the vessel is pressurized (or contains a static head of liquid acting at the closure) will not:

a) Cause or allow the closure to be opened or leak; or

b) Result in the failure of any other locking component or holding element; or

c) Increase the stress in any other locking component or holding element by more than 50% above the allowable stress of the component.

4.8.3.3 Quick-actuating closures shall be designed and installed such that it may be determined by visual external observation that the holding elements are in satisfactory condition.

4.8.3.4 Quick-actuating closures shall also be designed so that all locking components can be verified to be fully engaged by visual observation or other means prior to the application of pressure to the vessel.

4.8.3.5 When installed, all vessels having quick-actuating closures shall be provided with a pressure-indicating device visible from the operating area and suitable to detect pressure at the closure.

4.8.4 Specific Design Requirements

4.8.4.1 Quick-actuating closures that are held in position by positive locking devices and that are fully released by partial rotation or limited movement of the closure itself or the locking mechanism and any closure that is other than manually operated shall be so designed that when the vessel is installed the following conditions are met:

a) The closure and its holding elements are fully engaged in their intended operating position before pressure can be applied in the vessel.

b) Pressure tending to force the closure open or discharge the contents clear of the vessel shall be released before the closure can be fully opened for access.

c) In the event that compliance with paragraphs 4.8.4.1.a and 4.8.4.1.b above is not inherent in the design of the closure and its holding elements, provisions shall be made so that devices to accomplish this can be added when the vessel is installed.

4.8.4.2 The design rules of paragraph 4.16 of this code may not be applicable to design Quick-Actuating or Quick-Opening Closures, see paragraph 4.16.1.4.

4.8.4.3 The designer shall consider the effects of cyclic loading, other loadings (see paragraph 4.1.5.3) and mechanical wear on the holding and locking components.

4.8.4.4 It is recognized that it is impractical to write requirements to cover the multiplicity of devices used for quick access, or to prevent negligent operation or the circumventing of safety devices. Any device or devices that will provide the safeguards broadly described in paragraphs 4.8.4.1.a, 4.8.4.1.b and 4.8.4.1.c above will meet the intent of this Division.

4.8.5 Alternative Designs for Manually Operated Closures

4.8.5.1 Quick-actuating closures that are held in position by a locking mechanism designed for manual operation shall be designed such that if an attempt is made to open the closure when the vessel is under pressure, the closure will leak prior to full disengagement of the locking components and release of the closure. The design of the closure and vessel shall be such that any leakage shall be directed away from the normal position of the operator.

4.8.5.2 Manually operated closures need not satisfy paragraphs 4.8.4.1.a, 4.8.4.1.b and 4.8.4.1.c, but such closures shall be equipped with an audible or visible warning device that will warn the operator if pressure is applied to the vessel before the holding elements and locking components are fully engaged in their intended position or if an attempt is made to disengage the locking mechanism before the pressure within the vessel is released.

4.8.6 Supplementary Requirements for Quick-Actuating (Quick-Opening) Closures

Annex 4.B provides additional design information for the Manufacturer and provides installation, operational, and maintenance requirements for the Owner.

4.9 Design Rules for Braced and Stayed Surfaces

4.9.1 Scope

4.9.1.1 Design requirements for braced and stayed surfaces are provided in this paragraph. Requirements for the plate thickness and requirements for the staybolt or stay geometry including size, pitch, and attachment details are provided.

4.9.2 Required Thickness of Braced and Stayed Surfaces

4.9.2.1 The minimum thickness for braced and stayed flat plates and those parts that, by these rules, require staying as flat plates with braces or staybolts of uniform diameter symmetrically spaced, shall be calculated by the following equation.

$$t = p_s \sqrt{\frac{P}{SC}} \tag{4.9.1}$$

4.9.2.2 When stays are used to connect two plates, and only one of these plates requires staying, the value of C shall be governed by the thickness of the plate requiring staying.

4.9.3 Required Dimensions and Layout of Staybolts and Stays

4.9.3.1 The required area of a staybolt or stay at its minimum cross section, usually located at the root of the thread, exclusive of any corrosion allowance, shall be obtained by dividing the load on the staybolt computed in accordance with paragraph 4.9.3.2 by the allowable tensile stress value for the staybolt material, multiplying the result by 1.10.

4.9.3.2 The area supported by a staybolt or stay shall be computed on the basis of the full pitch dimensions, with a deduction for the area occupied by the stay. The load carried by a stay is the product of the area supported by the stay and the maximum allowable working pressure. When a staybolt or stay for a shell is unsymmetrical because of interference with other construction details, the area supported by the staybolt or stay shall be computed by taking the distance from the center of the spacing on one side of the staybolt or stay to the center of the spacing on the other side.

4.9.3.3 When the edge of a flat stayed plate is flanged, the distance from the center of the outermost stays to the inside of the supporting flange shall not be greater than the pitch of the stays plus the inside radius of the flange.

4.9.4 Requirements for Welded-in Staybolts and Welded Stays

4.9.4.1 Welded-in staybolts may be used provided the following requirements are satisfied.

a) The configuration is in accordance with the typical arrangements shown in Figure 4.9.1.

b) The required thickness of the plate shall not exceed 38 mm (1.5 in.).

c) The maximum pitch shall not exceed 15 times the diameter of the staybolt; however, if the required plate thickness is greater than 19 mm (0.75 in.), the staybolt pitch shall not exceed 508 mm (20 in.).

d) The size of the attachment welds is not less than that shown in Figure 4.9.1.

e) The allowable load on the welds shall not exceed the product of the weld area (based on the weld dimension parallel to the staybolt), the allowable tensile stress of the material being welded, and a weld joint factor of 60%.

4.9.4.2 Welded stays may be used provided the following requirements are satisfied.

a) The configuration is in accordance with the typical arrangements shown in Figure 4.9.1.

b) The pressure does not exceed 2 MPa (300 psi).

c) The required thickness of the plate does not exceed 13 mm (0.5 in.).

d) The size of the fillet welds is not less than the plate thickness requiring stay.

e) The inside welds are visually examined before the closing plates are attached.

f) The allowable load on the fillet welds shall not exceed the product of the weld area (based on the minimum leg dimension), the allowable tensile stress of the material being welded, and a weld joint factor of 55%.

g) The maximum diameter or width of the hole in the plate shall not exceed 32 mm (1.25 in.).

h) The maximum pitch, p_s, is determined by Equation (4.9.1) with $C = 2.1$ if either plate thickness is less than or equal to 11 mm (0.4375 in.) thick, and $C = 2.2$ for all other plate thicknesses.

4.9.5 Nomenclature

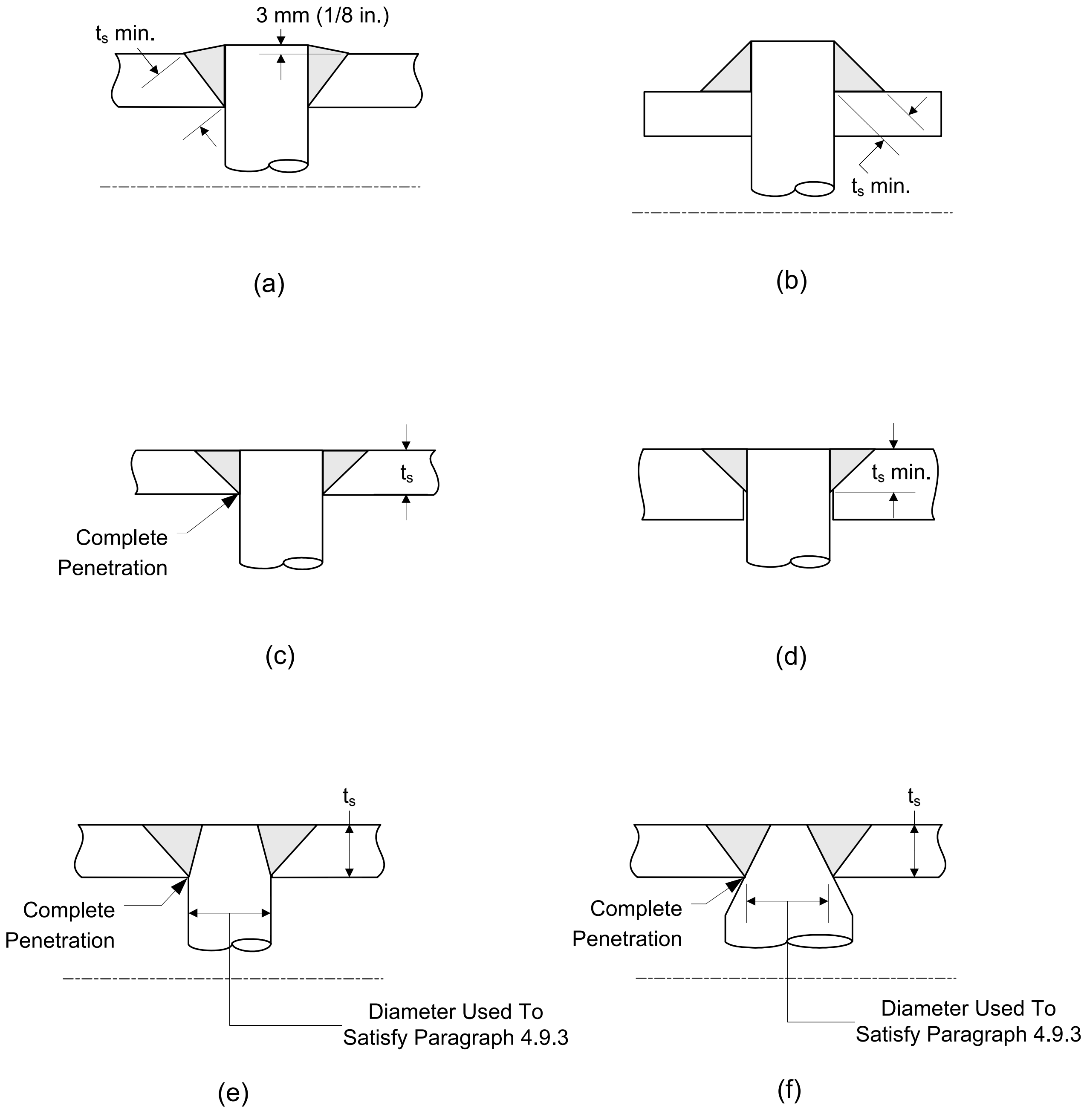
C	stress factor for braced and stayed surfaces (see Table 4.9.1).
P	design pressure.
p_s	maximum pitch. The maximum pitch is the greatest distance between any set of parallel straight lines passing through the centers of staybolts in adjacent rows. Each of the three parallel sets running in the horizontal, the vertical, and the inclined planes shall be considered.
S	allowable stress from Annex 3.A evaluated at the design temperature.
t	minimum required plate thickness.
t_s	nominal thickness of the thinner stayed plate (see Figure 4.9.1).

4.9.6 Tables

Table 4.9.1 – Stress Factor For Braced And Stayed Surfaces

Braced And Stayed Surface Construction	C
Welded stays through plates not over 11 mm (0.4375 in.) in thickness	2.1
Welded stays through plates over 11 mm (0.4375 in.) in thickness	2.2

4.9.7 Figures



Note: t_s is the nominal thickness of the thinner stayed plate

Figure 4.9.1 – Typical Forms of Welded Staybolts

4.10 Design Rules for Ligaments

4.10.1 Scope

4.10.1.1 Rules for determining the ligament efficiency for hole patterns in cylindrical shells are covered in this paragraph. The ligament efficiency or weld joint factor (see paragraph 4.10.3) is used in conjunction with the design equations for shells in paragraph 4.3.

4.10.2 Ligament Efficiency

4.10.2.1 When a cylindrical shell is drilled for tubes in a line parallel to the axis of the shell for substantially the full length of the shell as shown in Figures 4.10.1 through 4.10.3, the efficiency of the ligaments between the tube holes shall be determined as follows.

a) When the pitch of the tube holes on every row is equal (see Figure 4.10.1), the ligament efficiency is:

$$E = \frac{p-d}{p} \tag{4.10.1}$$

b) When the pitch of tube holes on any one row is unequal (as in Figures 4.10.2 and 4.10.3), the ligament efficiency is:

$$E = \frac{p_1 - nd}{p_1} \tag{4.10.2}$$

c) When the adjacent longitudinal rows are drilled as described in paragraph 4.10.2.1.b, diagonal and circumferential ligaments shall also be examined. The least equivalent longitudinal efficiency shall be used to determine the minimum required thickness and the maximum allowable working pressure.

d) When a cylindrical shell is drilled for holes so as to form diagonal ligaments, as shown in Figure 4.10.4, the efficiency of these ligaments shall be determined by Figures 4.10.5 and 4.10.6. Figure 4.10.5 is used to determine the efficiency of longitudinal and diagonal ligaments with limiting boundaries where the condition of equal efficiency of diagonal and longitudinal ligaments form one boundary and the condition of equal efficiency of diagonal and circumferential ligaments form the other boundary. Figure 4.10.6 is used for determining the equivalent longitudinal efficiency of diagonal ligaments. This efficiency is used in the equations for setting the minimum required thickness.

   1) Figure 4.10.5 is used when either or both longitudinal and circumferential ligaments exist with diagonal ligaments. To use Figure 4.10.5, compute the value of p^*/p_1 and also the efficiency of the longitudinal ligament. Next find in the diagram, the vertical line corresponding to the longitudinal efficiency of the ligament and follow this line vertically to the point where it intersects the diagonal line representing the ratio of p^*/p_1. Then project this point horizontally to the left, and read the diagonal efficiency of the ligament on the scale at the edge of the diagram. The minimum shell thickness and the maximum allowable working pressure shall be based on the ligament that has the lower efficiency.

   2) Figure 4.10.6 is used for holes that are not in-line, or holes that are placed longitudinally along a cylindrical shell. The diagram may be used for pairs of holes for all planes between the longitudinal plane and the circumferential plane. To use Figure 4.10.6, determine the angle θ between the longitudinal shell axis and the line between the centers of the openings and compute the value of p^*/d. Find in the diagram, the vertical line corresponding to the value of θ and follow this line vertically to the line representing the value of p^*/d. Then project this point horizontally to the left, and read the equivalent longitudinal efficiency of the diagonal ligament. This equivalent longitudinal

efficiency is used to determine the minimum required thickness and the maximum allowable working pressure.

e) When tube holes in a cylindrical shell are arranged in symmetrical groups which extend a distance greater than the inside diameter of the shell along lines parallel to the axis and the same spacing is used for each group, the efficiency for one of the groups shall be not less than the efficiency on which the maximum allowable working pressure is based.

f) The average ligament efficiency in a cylindrical shell, in which the tube holes are arranged along lines parallel to the axis with either uniform or non-uniform spacing, shall be computed by the following rules and shall satisfy the requirements of both. These rules only apply to ligaments between tube holes and not to single openings. They may give lower efficiencies in some cases than those for symmetrical groups which extend a distance greater than the inside diameter of the shell as covered in paragraph 4.10.2.1.e. When this occurs, the efficiencies computed by the rules under paragraph 4.10.2.1.b shall govern.

   1) For a length equal to the inside diameter of the shell for the position which gives the minimum efficiency, the efficiency shall be not less than that on which the maximum allowable working pressure is based. When the inside diameter of the shell exceeds 1520 mm (60 in.), the length shall be taken as 1520 mm (60 in.), in applying this rule.

   2) For a length equal to the inside radius of the shell for the position which gives the minimum efficiency, the efficiency shall be not less than 80% of that on which the maximum allowable working pressure is based. When the inside radius of the shell exceeds 762 mm (30 in.), the length shall be taken as 762 mm (30 in.), in applying this rule.

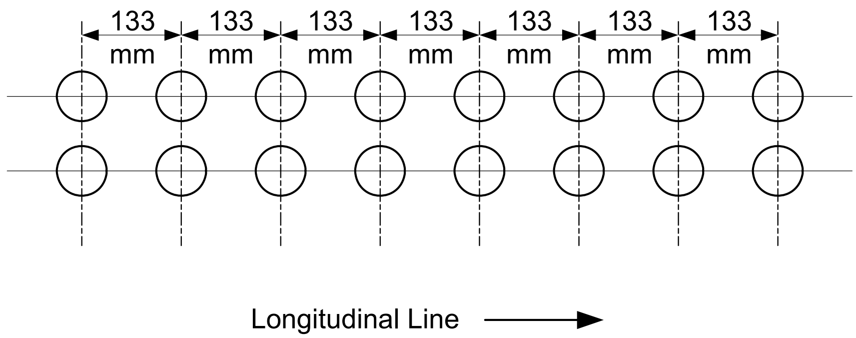
4.10.3 Ligament Efficiency and the Weld Joint Factor

When ligaments occur in cylindrical shells made from welded pipe or tubes and their calculated efficiency is less than 85% (longitudinal) or 50% (circumferential), the efficiency to be used in paragraph 4.3 to determine the minimum required thickness is the calculated ligament efficiency. In this case, the appropriate stress value in tension may be multiplied by the factor 1.18.

4.10.4 Nomenclature

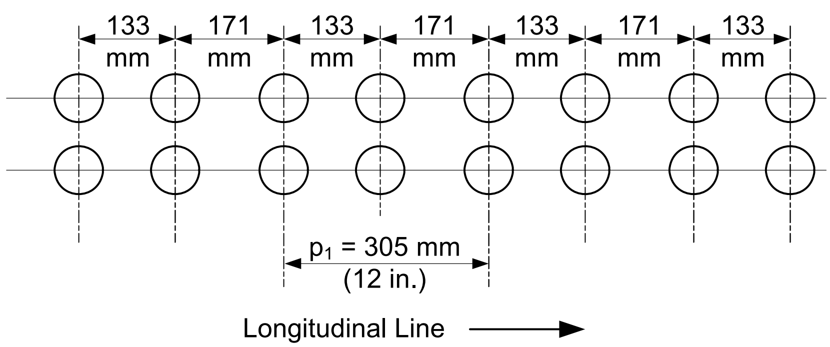
d	diameter of tube holes
E	longitudinal ligament efficiency
E_{long}	longitudinal ligament efficiency in percent
p	longitudinal pitch of tube holes
p_1	unit length of ligament
$p*$	diagonal pitch of tube holes
θ	angle of the diagonal pitch with respect to the longitudinal line
s	longitudinal dimension of diagonal pitch, $p*\cdot\cos\theta$
n	number of tube holes in length p_1

4.10.5 Figures



General Note: 133 mm (5-1/4 in.)

Figure 4.10.1 – Example of Tube Spacing With the Pitch of Holes Equal in Every Row



General Note: 133 mm (5-1/4 in.)
171 mm (6-3/4 in.)

Figure 4.10.2 – Example of Tube Spacing With the Pitch of Holes Unequal in Every Second Row

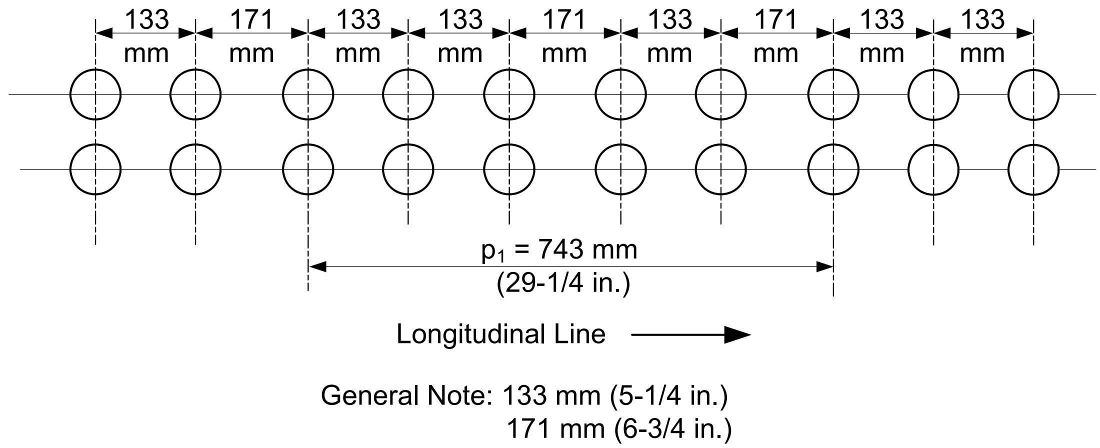


Figure 4.10.3 – Example of Tube Spacing With the Pitch of Holes Varying in Every Second and Third Row

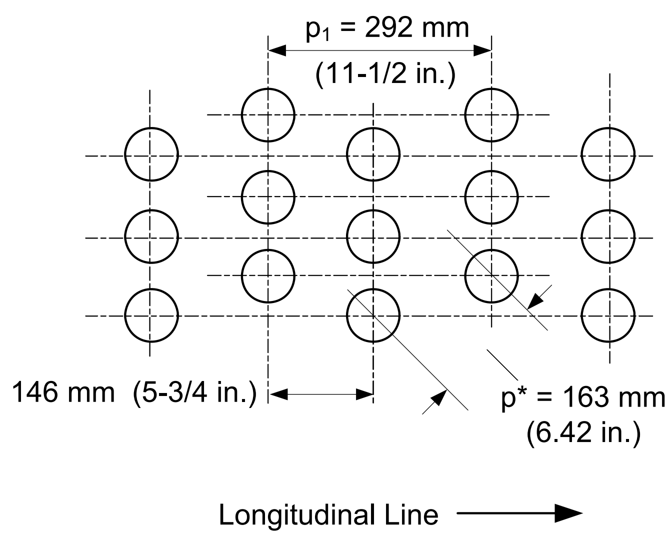
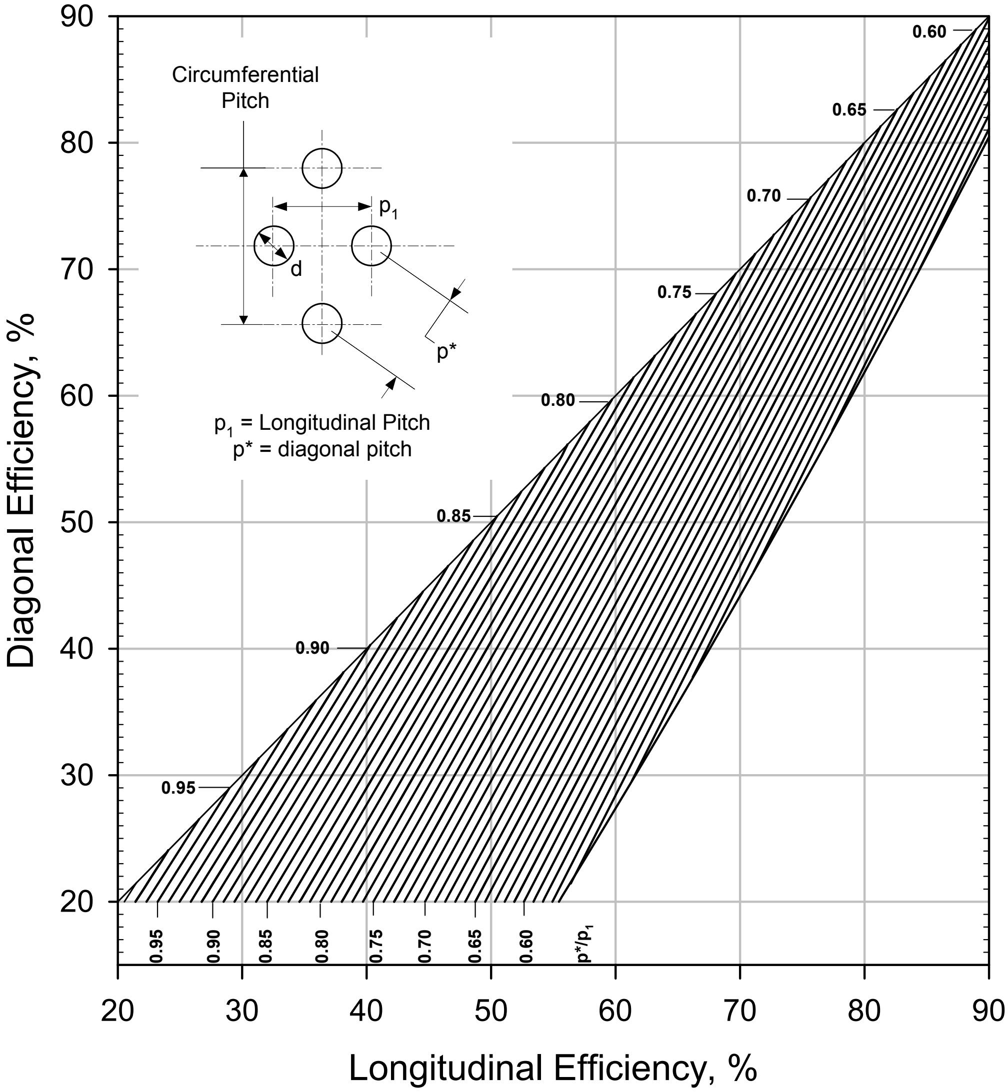


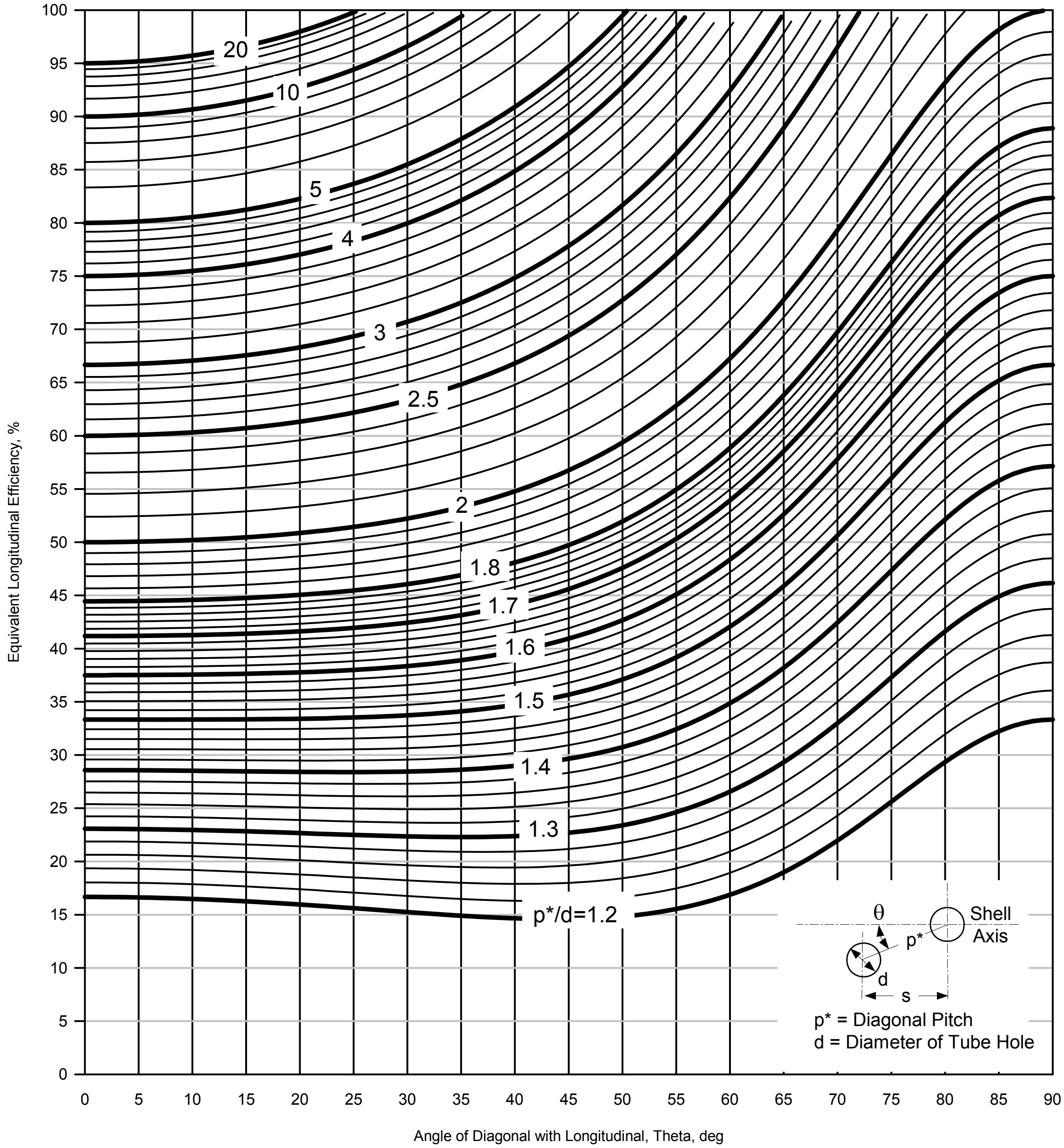
Figure 4.10.4 – Example of Tube Spacing With the Tube Holes on Diagonal Lines



Notes:

1. Equations are provided for the curve shown above, use of these equations is permitted for values beyond the values shown in this curve.
2. Diagonal efficiency: $\% = \dfrac{J + 0.25 - \left(1 - 0.01 \cdot E_{long}\right)\sqrt{0.75 + J}}{0.00375 + 0.005 \cdot J}, \quad where\ \ J = \left(\dfrac{p^*}{p_1}\right)^2$
3. The curve of condition of equal efficiency of diagonal and circumferential ligaments is given by:
$$\% = \frac{200M + 100 - 2\left(100 - E_{long}\right)\sqrt{1+M}}{1+M}, \quad where\ \ M = \left(\frac{100 - E_{long}}{200 - 0.5E_{long}}\right)^2$$
4. $E_{long} = 100\left(\dfrac{p_1 - d}{p_1}\right)$

Figure 4.10.5 – Diagram for Determining the Efficiency of Longitudinal and Diagonal Ligaments Between Openings in Cylindrical Shells



Notes:

1. An equation is provided for the curve shown above, the use of this equation is permitted for values beyond the values shown in this curve.
2. Equivalent longitudinal efficiency: $$\% = \frac{\sec^2\theta + 1 - \left(\dfrac{\sec\theta}{p^*/d}\right)\sqrt{3+\sec^2\theta}}{0.015 + 0.005\sec^2\theta}$$

Figure 4.10.6 – Diagram for Determining the Equivalent Efficiency of Diagonal Ligaments Between Openings in Cylindrical Shells

4.11 Design Rules for Jacketed Vessels

4.11.1 Scope

4.11.1.1 The minimum requirements for the design of the jacketed portion of a pressure vessel shall conform to the requirements given in paragraph 4.11. The jacketed portion of the vessel is defined as the inner and outer walls, the closure devices and all other penetration or parts within the jacket that are subjected to pressure stress. Parts such as nozzle closure members and stay rings are included in this definition. For the purposes of this section, jackets are assumed to be integral pressure chambers, attached to a vessel for one or more purposes, such as:

a) To heat the vessel and its contents,

b) To cool the vessel and its contents, or

c) To provide a sealed insulation chamber for the vessel.

4.11.1.2 Paragraph 4.11 applies only to jacketed vessels having jackets over the shell or heads as illustrated in Figure 4.11.1, partial jackets as illustrated in Figure 4.11.2, and half–pipe jackets as illustrated in Figure 4.11.3.

4.11.1.3 The jacketed vessels shown in Figure 4.11.1 are categorized as five types shown below. For these types of vessels, the jackets shall be continuous circumferentially for Types 1, 2, 4 or 5 and shall be circular in cross section for Type 3. The use of any combination of the types shown is permitted on a single vessel provided the individual requirements for each are met. Nozzles or other openings in Type 1, 2, 4 or 5 jackets that also penetrate the vessel shell or head shall be designed in accordance with paragraph 4.5. Paragraph 4.11 does not cover dimpled or embossed jackets.

a) Type 1 – Jacket of any length confined entirely to the cylindrical shell

b) Type 2 – Jacket covering a portion of the cylindrical shell and one head

c) Type 3 – Jacket covering a portion of one head

d) Type 4 – Jacket with addition of stay or equalizer rings to the cylindrical shell portion to reduce the effective length

e) Type 5 – Jacket covering the cylindrical shell and any portion of either head.

4.11.1.4 Paragraph 4.11 does not contain rules to cover all details of design and construction. Jacket types defined in paragraph 4.11.1.3 subject to general loading conditions (i.e. thermal gradients) or jacket types of different configurations subject to general loading conditions shall be designed using Part 5.

4.11.1.5 If the internal pressure is 100 kPa (15 psi) or less, and any combination of pressures and vacuum in the vessel and jacket will produce a total pressure greater than 100 kPa (15 psi) on the inner vessel wall, then the entire jacket is within the scope of paragraph 4.11.

4.11.2 Design of Jacketed Shells and Jacketed Heads

4.11.2.1 Shell and head thickness shall be determined using paragraphs 4.3 and 4.4 as applicable. In consideration of the loadings given in paragraph 4.1, particular attention shall be given to the effects of local internal and external pressure loads and differential thermal expansion (see paragraph 4.11.1.4). Where vessel supports are attached to the jacket, consideration shall be given to the transfer of the supported load of the inner vessel and contents.

4.11.2.2 The requirements for inspection openings in jackets shall be in accordance with paragraph 4.5.16 except that the maximum size of inspection openings in the jacketed portion of the vessel need not exceed DN 50 (NPS 2) pipe for all diameter vessels.

4.11.2.3 The use of impingement plates or baffles at the jacket inlet connection to reduce erosion of the inner wall shall be considered for media where vapors are condensed (i.e. steam).

4.11.2.4 Flat plate regions of jacketed vessels may be designed as braced and stayed surfaces using the rules of paragraph 4.9.

4.11.3 Design of Closure Member of Jacket to Vessel

4.11.3.1 The design of jacket closure members shall be in accordance with Table 4.11.1 and the additional requirements of paragraph 4.11.3. Alternative geometries to those illustrated may be used in accordance with paragraph 4.11.1.4.

4.11.3.2 Any radial welds in closure members shall be butt-welded joints penetrating through the full thickness of the member and shall be ground flush where attachment welds are to be made.

4.11.3.3 Partial penetration and fillet welds are permitted when both of the following requirements are satisfied.

a) The material of construction satisfies the following equation.

$$\frac{S_{yT}}{S_u} \le 0.625 \tag{4.11.1}$$

b) The component is not in cyclic service, i.e. a fatigue analysis is not required in accordance with paragraph 4.1.1.4.

4.11.3.4 Closures for any type of stay-bolted jacket may be designed in accordance with the requirements of Type 1 jackets shown in Figure 4.11.1 provided the entire jacket is stay-bolted to compensate for pressure end forces.

4.11.4 Design of Penetrations Through Jackets

4.11.4.1 The design of openings through the jacket space shall be in accordance with the rules given in paragraph 4.5. Reinforcement of the opening in the jacket shall not be required for penetrations of the type shown in Table 4.11.2 since the opening is stayed by virtue of the nozzle or neck of the closure member.

4.11.4.2 Jacket penetration closure member designs shown in Table 4.11.2 shall conform to the following requirements stipulated in this table and the following provisions. Alternative geometries to those illustrated may be used if the design is based on Part 5.

a) The jacket penetration closure member minimum thickness considers only pressure membrane loading. Axial pressure loadings and secondary loadings given in paragraph 4.1 shall be considered in the design.

b) The design Details 2, 3, 4, 5 and 6 shown in Table 4.11.2 provide some flexibility. Only pressure membrane loading is considered in establishing the minimum thickness of the penetration closure member. If the localized stresses at the penetration detail need to be established, the methodology in Part 5 shall be used.

c) All radial welds in opening sealer membranes shall be butt-welded joints that penetrate through the full thickness of the member.

d) Closure member welds shall be circular, elliptical or obround in shape where possible. Rectangular member welds are permissible provided that corners are rounded to a suitable radius.

e) The requirements of paragraph 4.11.3.3 shall be satisfied.

4.11.5 Design of Partial Jackets

4.11.5.1 Partial jackets include jackets that encompass less than the full circumference of the vessel. Some variations are shown in Figure 4.11.2.

4.11.5.2 The rules for construction of jacketed vessels in the preceding paragraphs also apply to partial jackets, with the following exceptions.

a) Stayed partial jackets shall be designed and constructed in accordance with paragraph 4.9 with closures designed in accordance with paragraph 4.11.3.

b) Partial jackets that, by virtue of their service or configuration, do not lend themselves to staybolt construction may be fabricated by other means provided they are designed using Part 5.

4.11.6 Design of Half–Pipe Jackets

4.11.6.1 The rules in this section are applicable for the design of half-pipe jackets constructed of NPS 2, 3 or 4 pipes and subjected to internal pressure loading (see Figure 4.11.3). Configurations that do not satisfy the rules in paragraph 4.11.6.1 may be designed in accordance with Part 5.

4.11.6.2 The fillet weld attaching the half-pipe jacket to the vessel shall have a throat thickness not less than the smaller of the jacket or shell thickness. Consideration should be given to the selection of the half-pipe jacket pitch needed to provide welder access. In addition, the requirements of paragraph 4.11.3.3 shall be satisfied.

4.11.6.3 The minimum required thickness of a half pipe jacket is given by the following equation. For a design to be acceptable, the additional condition that $P_j \le P_{jpm}$ where P_{jpm} is given by Equation (4.11.3) must also be satisfied.

$$t_{rp} = \frac{P_j r_p}{0.85S_j - 0.6P_j} \tag{4.11.2}$$

4.11.6.4 The maximum permissible pressure in the half-pipe jacket, P_{jpm}, shall be determine using the following equation.

$$P_{jpm} = \frac{F_p}{K_p} \tag{4.11.3}$$

where

$$F_p = \min\left[\left(1.5S - S^*\right),\ 1.5S\right] \tag{4.11.4}$$

$$K_p = C_1 + C_2 D^{0.5} + C_3 D + C_4 D^{1.5} + C_5 D^2 + C_6 D^{2.5} + C_7 D^3 + C_8 D^{3.5} + C_9 D^4 + C_{10} D^{4.5} \tag{4.11.5}$$

The coefficients for Equation (4.11.5) are provided in Table 4.11.3.

4.11.7 Nomenclature

D inside diameter of the inner vessel.
D_{pj} nominal pipe size of the half-pipe jacket.
K_p half-pipe jacket rating factor.
P_j design pressure in the jacket chamber.
P_{jpm} permissible jacket pressure based on the jacket and shell geometry.
j jacket space defined as the inside radius of the jacket minus the outside radius of the inner vessel.
L length of the jacket.
t_c nominal thickness of the closure member.
t_j nominal thickness of the outer jacket wall.
t_n nominal thickness of the nozzle.
t_s nominal thickness of the shell inner wall.
t_{rj} required minimum thickness of the outer jacket wall.
t_{rc} required minimum thickness of the closure member.
t_{rp} required minimum thickness of the half-pipe jacket.
R_j inside radius of the jacket.
R_p radius of the opening in the jacket at the jacket penetration
R_s outside radius of the inner vessel.
r corner radius of torus closures.
r_p inside radius of the half-pipe jacket.
S allowable stress of the inner shell from Annex 3.A at the design temperature.
S_c allowable stress of the jacket closure from Annex 3.A at the design temperature.
S_j allowable stress of the jacket from Annex 3.A at the design temperature.
S_{yT} yield strength from Annex 3.D at the design temperature.
S_u minimum specified ultimate tensile strength from Annex 3.D.
S^* actual longitudinal tensile stress in the head or shell due to internal pressure and other axial forces; when axial forces are negligible, $S^* = PD/4t_s$. If the combination of axial forces and pressure results in a negative value of S^*, then $S^* = 0$.

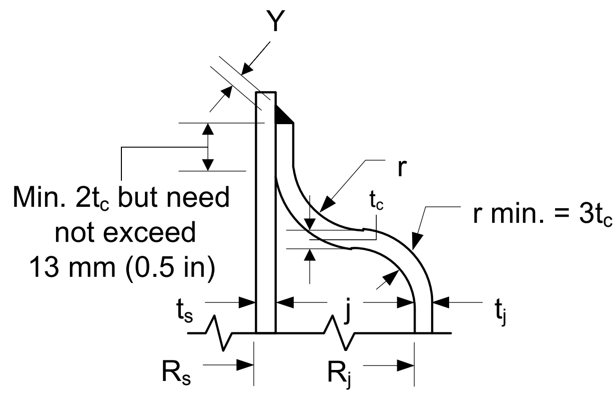
4.11.8 Tables

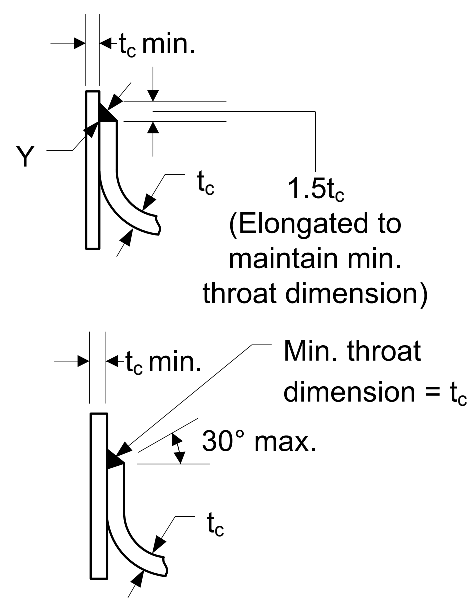
Table 4.11.1 – Design Of Closure Member Of Jacket To Shell

<table>
<tr><th>Detail</th><th>Requirements</th><th>Figure</th></tr>
<tr><td>1</td><td>Closure details (a) and (b) shall only be used when the requirements of paragraph 4.11.3.3 are satisfied.

These closures may be used on Types 1, 2, and 4 jacketed vessels as shown in Figure 4.11.1 and shall have t_{rc} of at least equal to t_{rj} and corner radius r shall not be less than $3t_c$.

These closure designs are limited to a maximum thickness t_{rc} of 16 mm (0.625 in.)

When this construction is used on Type 1 jacketed vessels, the weld dimension Y shall be not less than $0.7t_c$; and when used on Type 2 and 4 jacketed vessels, the weld dimension Y shall be not less than $0.83t_c$.</td><td>


(a) Type 1 Jackets


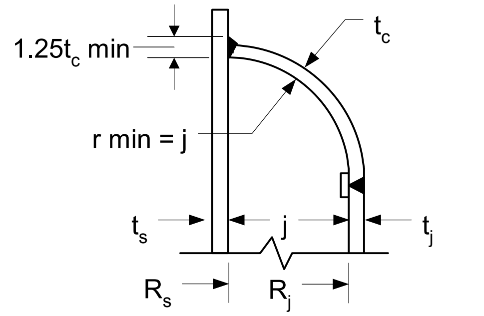
(b) Types 2 and 4 Jackets</td></tr>
</table>

Table 4.11.1 – Design Of Closure Member Of Jacket To Shell

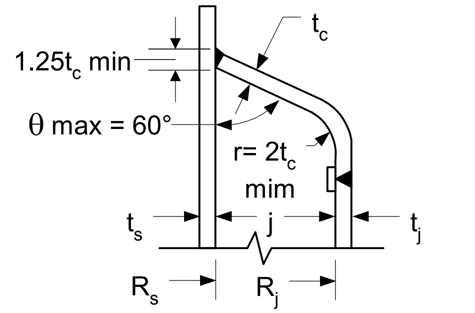
<table>
<tr><th>Detail</th><th>Requirements</th><th>Figure</th></tr>
<tr><td>2</td><td>These closures shall have t_{rc} at least equal to t_{rj}. In addition for Detail (c), t_{rc} shall be not less than the following:

$$t_{rc} = 0.707 j \sqrt{\frac{P_j}{S_c}}$$

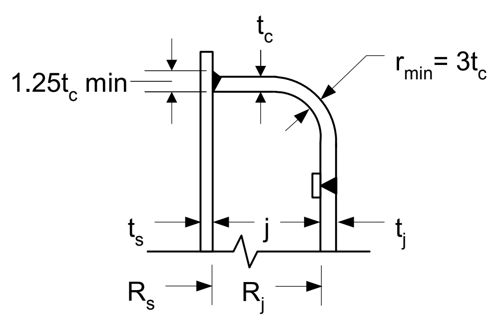
A groove weld attaching the closure to the inner vessel and fully penetrating the closure thickness t_c may be used with any of the types of jacketed vessels shown in Figure 4.11.1. However, a fillet weld having a minimum throat dimension of $0.7t_c$ may also be used to join the closure of the inner vessel on Type 1 jacketed vessels of Figure 4.11.1.

The closure and jacket shell may be one-piece construction or welded using a full penetration butt weld. A backing strip may be used.</td><td>

(a)



(b)



(c)</td></tr>
<tr><td>3</td><td>This closure shall be used only on Type 1 jacketed vessels shown in Figure 4.11.1.

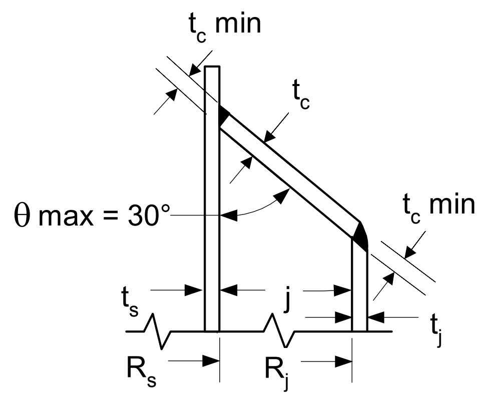
The closure thickness t_{rc} shall be computed using the Equation for a conical shell in paragraph 4.3, but shall be not less than t_{rj}. The angle θ shall be limited to 30 degrees maximum.</td><td>
</td></tr>
</table>

Table 4.11.1 – Design Of Closure Member Of Jacket To Shell

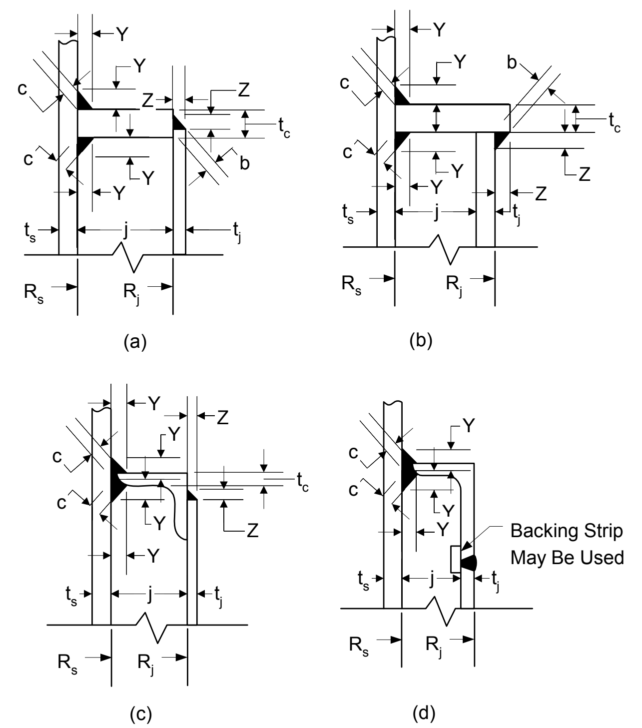
Detail	Requirements	Figure
4	Closure details (a), (b), and (c) shall only be used when the requirements of paragraph 4.11.3.3 are satisfied. These closures shall be used only on Type 1 jacketed vessels as shown in Figure 4.11.1 and with the further limitation that t_{rj} does not exceed 16 mm (0.625 in.). The required minimum thickness for the closure bar shall be equal to: $t_{rc} = \max\left[2t_{rj},\ \left(0.707\, j\sqrt{\frac{P_j}{S_c}}\right)\right]$ Fillet weld sizes shall be as follows: $Y \geq \min\left[0.75t_c,\ 0.75t_s\right]$ and $c = 0.7Y \ \min$ $Z \geq t_j$ and $b = 0.7Z \ \min$	

Table 4.11.1 – Design Of Closure Member Of Jacket To Shell

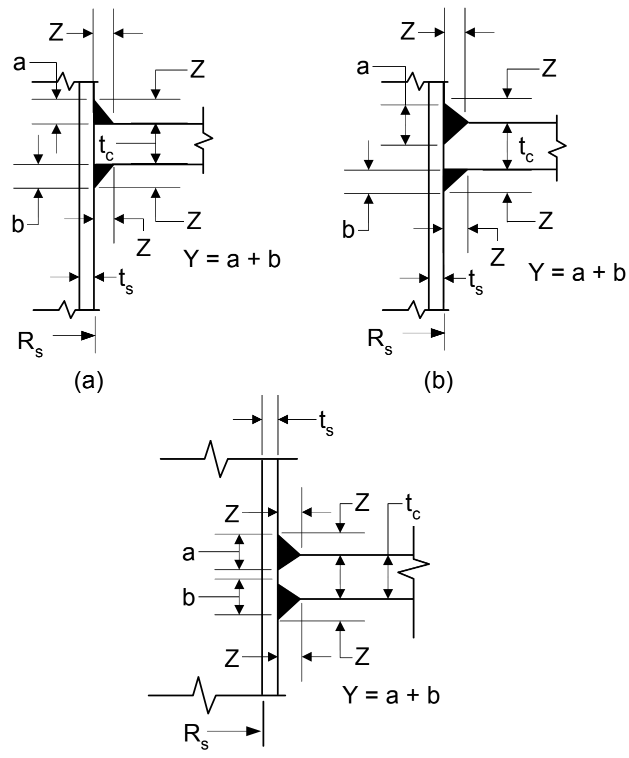
Detail	Requirements	Figure
5	Closure details (a), (b), and (c) shall only be used when the requirements of paragraph 4.11.3.3 are satisfied. These closures may be used on any of the types of jacketed vessels shown in Figure 4.11.1. For Type 1 jacketed vessels, the required minimum closure bar thickness shall be determined from the equations in Table 4.11.1, Detail 4. For all other types of jacketed vessels, the required minimum closure bar thickness and the maximum allowable width of the jacket space shall be determined from the following equations: $t_{rc} = 1.414\sqrt{\left(\frac{P_j R_s j}{S_c}\right)}$ $j = \frac{2S_c t_s^2}{P_j R_j} - \frac{(t_s + t_j)}{2}$ Weld sizes connecting the closure bar to the inner vessel shall be as follows: $Y \geq \min[1.5t_c,\ 1.5t_s]$, and shall be measured as the sum of dimensions a and b as shown. Z is equal to the minimum fillet size necessary when used in conjunction with a groove weld or another fillet weld to maintain the minimum required Y dimension.	(a) (b) (c)

Table 4.11.1 – Design Of Closure Member Of Jacket To Shell

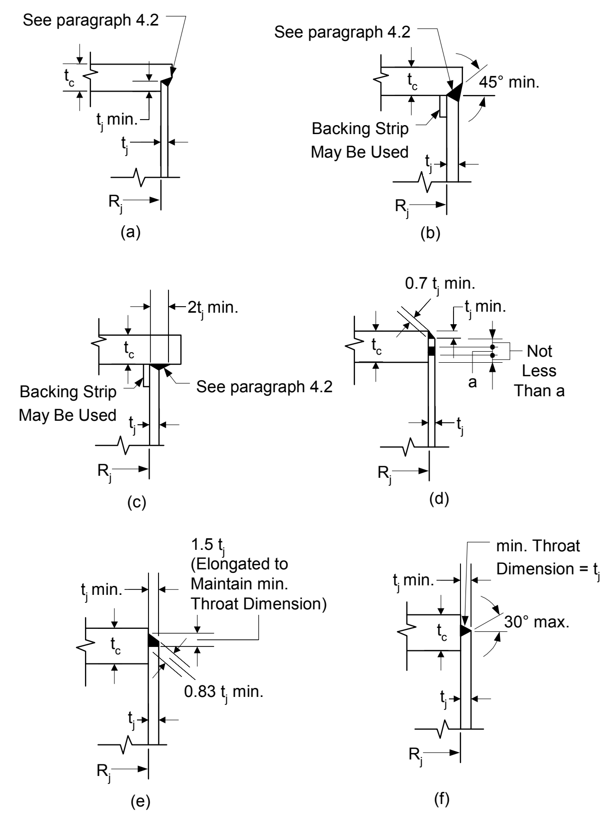
Detail	Requirements	Figure
6	Closure details (a), (b), and (c) shall only be used when the requirements of paragraph 4.11.3.3 are satisfied. The jacket to closure bar attachment welds shown in Details (a), (b) and (c) may be used on any of the types of jacketed vessels shown in Figure 4.11.1. Attachment welds shown in Details (d), (e) and (f) may be used on any of the types of jacketed vessels shown in Figure 4.11.1 where t_{rj} does not exceed 16 mm (0.625 in.). The required minimum closure bar thickness and the maximum allowable width of the jacket space shall be determined from the following equations: $t_{rc} = 1.414\sqrt{\left(\frac{P_j R_s j}{S_c}\right)}$ $j = \frac{2S_c t_s^2}{P_j R_j} - \frac{\left(t_s + t_j\right)}{2}$	

Table 4.11.1 – Design Of Closure Member Of Jacket To Shell

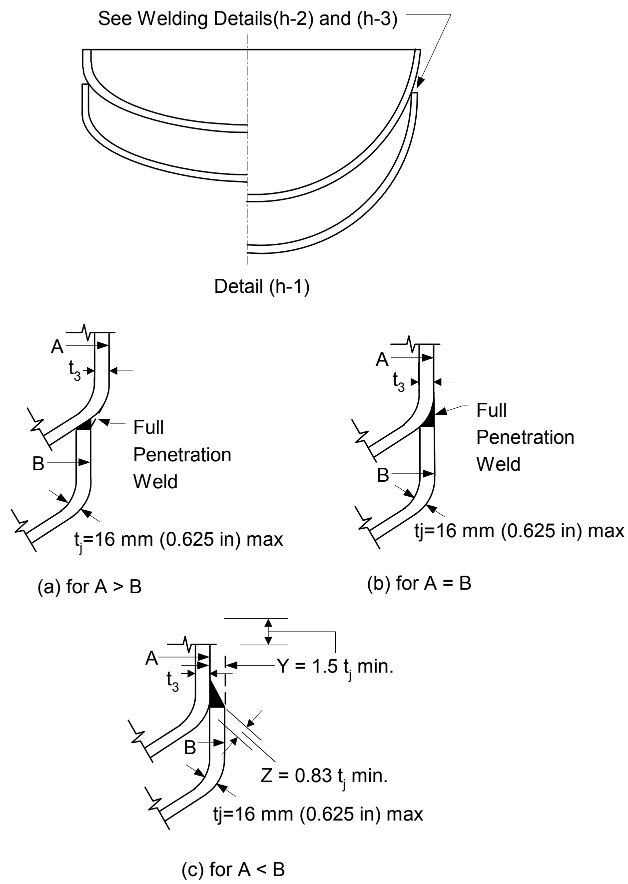
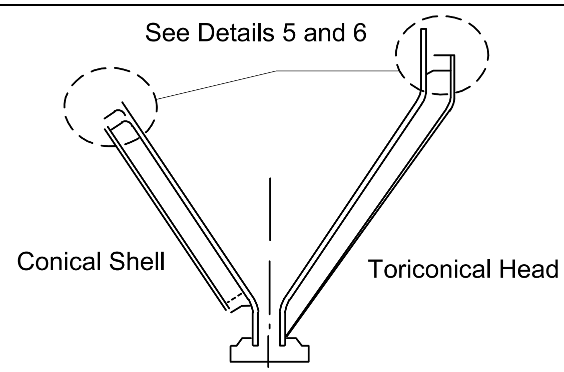
Detail	Requirements	Figure
7	Closure details (a), (b), and (c) shall only be used when the requirements of paragraph 4.11.3.3 are satisfied. These closures may be used on Type 3 jacketed vessels shown in Figure 4.11.1 shall have attachment welds in accordance with Details (a), (b) or (c). This construction is limited to jackets where t_{rj} does not exceed 16 mm (0.625 in.). For torispherical, ellipsoidal, and hemispherical heads, the outside diameter of jacket head shall not be greater than the outside diameter of the vessel head, or the inside diameter of the jacket head shall be nominally equal to the outside diameter of vessel head.	
8	Closures for conical or toriconical jackets shall comply with the requirements for Type 2 jacketed vessels shown in Figure 4.11.1.	

Table 4.11.2 – Design Of Jacket Penetration Details

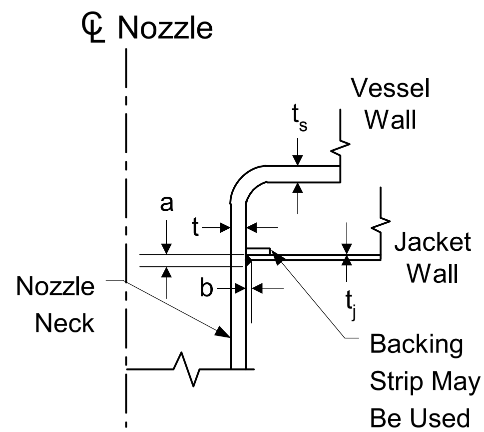
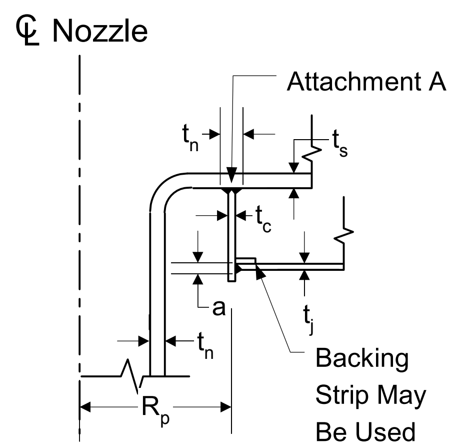
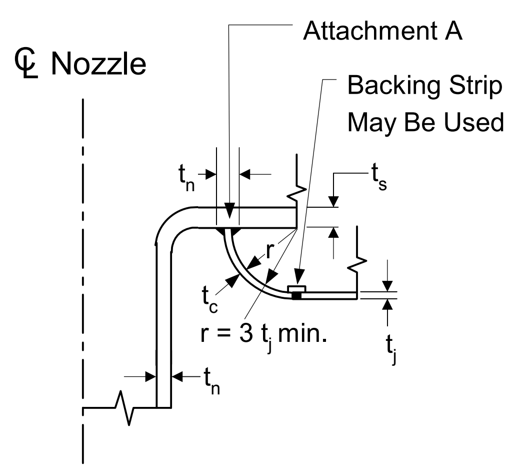
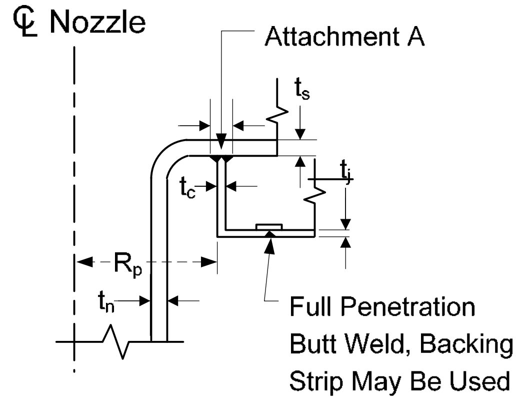
Detail	Requirements	Figure
1	This closure details shall only be used when the requirements of paragraph 4.11.3.3 are satisfied. The nozzle wall may be used as the closure member where the jacket is welded to the nozzle. $a = 2t_j \text{ min}$ and $b = t_j \text{ min}$	
2	This closure details shall only be used when the requirements of paragraph 4.11.3.3 are satisfied. The minimum required thickness, t_{rc}, for the geometries shall be calculated as a shell under external pressure in accordance with paragraph 4.4. $a = 2t_j \text{ min}$ and $b = t_j \text{ min}$ Attachment A shall be made using details in Table 4.2.6.	
3	This closure details shall only be used when the requirements of paragraph 4.11.3.3 are satisfied. The minimum required thickness, t_{rc}, shall be equal to t_{rj}. Attachment A shall be made using details in Table 4.2.6.	

Table 4.11.2 – Design Of Jacket Penetration Details

<table>
<tr><th>Detail</th><th>Requirements</th><th>Figure</th></tr>
<tr><td>4</td><td>This closure details shall only be used when the requirements of paragraph 4.11.3.3 are satisfied.

The minimum required thickness, t_{rc}, shall be calculated as a shell under external pressure in accordance with paragraph 4.4.

Attachment A shall be made using details in Table 4.2.6.</td><td></td></tr>
<tr><td>5</td><td>This closure details shall only be used when the requirements of paragraph 4.11.3.3 are satisfied.

The thickness required of the closure member attached to the inner vessel, t_{rc1}, shall be calculated as a shell under external pressure in accordance with paragraph 4.4. The required thickness of the flexible member, t_{rc2}, shall be determined as follows:

When a tubular section does not exist between jacket and torus:
$$t_{rc2} = \frac{Pr}{S_c E - 0.6P_j}$$
When a tubular section does exist between jacket and torus:
$$t_{rc2} = \frac{P_j R_p}{S_c E - 0.6P_j}$$
$a = 2t_j$, $b = t_j$, and $c = 1.25t_{c1}$

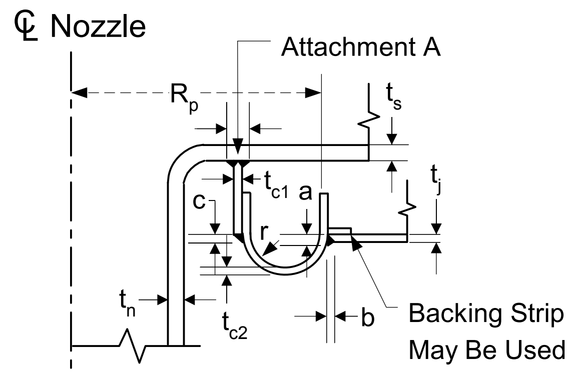
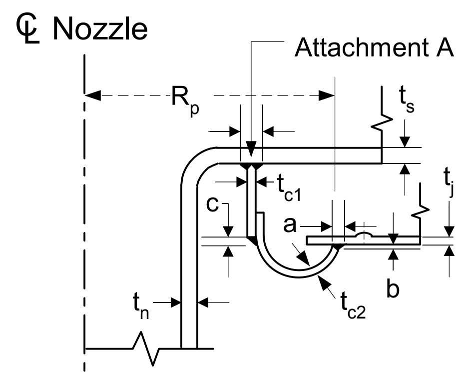
Attachment A shall be made using details in Table 4.2.6.</td><td>
(a)

(b)</td></tr>
</table>

Table 4.11.2 – Design Of Jacket Penetration Details

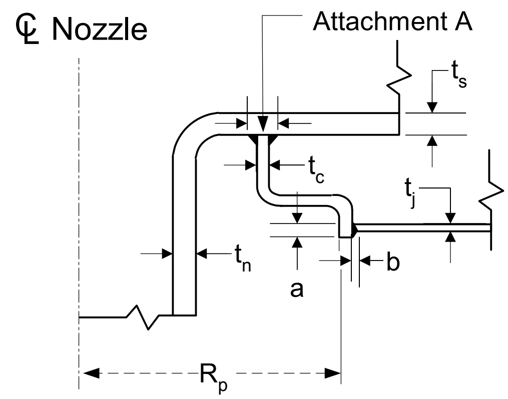
Detail	Requirements	Figure
6	This closure detail shall only be used when the requirements of paragraph 4.11.3.3 are satisfied. The minimum thickness, t_{rc}, shall be calculated as a shell of radius R_p under external pressure in accordance with paragraph 4.4. $a = 2t_j$ and $b = t_j$ Attachment A shall be made using details in Table 4.2.6.	

Table 4.11.3 – Coefficients For Equation (4.11.5)

D_{pj}	Coefficients	Shell Thickness			
		5 mm (3/16 in.)	6 mm (1/4 in.)	10 mm (3/8 in.)	13 mm (1/2 in.)
DN50 NPS 2	C_1	-3.6674510E+01	-1.8874043E+04	4.0083779E+02	-2.6447784E+02
	C_2	1.2306994E+01	1.7869518E+04	-5.7029108E+02	1.8066952E+02
	C_3	3.5701684E+00	-7.2846419E+03	3.1989698E+02	-4.9294965E+01
	C_4	-7.9516583E-01	1.6723763E+03	-9.4286208E+01	7.1522422E+00
	C_5	5.8791041E-02	-2.3648930E+02	1.6391764E+01	-5.7900069E-01
	C_6	-1.5365397E-03	2.1101742E+01	-1.7431218E+00	2.4758486E-02
	C_7	0.0000000E+00	-1.1608890E+00	1.1160179E-01	-4.3667599E-04
	C_8	0.0000000E+00	3.6022711E-02	-3.9549592E-03	0.0000000E+00
	C_9	0.0000000E+00	-4.8303253E-04	5.9644209E-05	0.0000000E+00
	C_{10}	0.0000000E+00	0.0000000E+00	0.0000000E+00	0.0000000E+00
DN80 NPS 3	C_1	-3.7588705E+03	-1.2551406E+04	-3.8104460E+04	-1.4263782E+04
	C_2	2.9919870E+03	1.2149900E+04	4.0491537E+04	1.6228077E+04
	C_3	-9.4177823E+02	-5.0657776E+03	-1.8844078E+04	-8.0227888E+03
	C_4	1.5278500E+02	1.1910361E+03	5.0415301E+03	2.2676555E+03
	C_5	-1.3452359E+01	-1.7255075E+02	-8.5435371E+02	-4.0440980E+02
	C_6	6.1167422E-01	1.5770136E+01	9.5115501E+01	4.7257835E+01
	C_7	-1.1235632E-02	-8.8782173E-01	-6.9588768E+00	-3.6233229E+00
	C_8	-2.1465752E-06	2.8148933E-02	3.2277515E-01	1.7597455E-01
	C_9	0.0000000E+00	-3.8488963E-04	-8.6172557E-03	-4.9179021E-03
	C_{10}	0.0000000E+00	0.0000000E+00	1.0094910E-04	6.0315412E-05
DN100 NPS 4	C_1	-2.1336346E+04	7.3995872E+03	8.3115447E+02	-4.0097574E+02
	C_2	1.5982068E+04	-6.7592710E+03	-7.6253222E+02	4.2602525E+02
	C_3	-4.9936486E+03	2.6131811E+03	2.9500674E+02	-1.7446665E+02
	C_4	8.4914220E+02	-5.4873257E+02	-6.1135935E+01	3.7753845E+01
	C_5	-8.4931392E+01	6.7571708E+01	7.4233181E+00	-4.6748939E+00
	C_6	5.0044853E+00	-4.8769663E+00	-5.2938127E-01	3.3376011E-01
	C_7	-1.6105634E-01	1.9112909E-01	2.0558271E-02	-1.2795569E-02
	C_8	2.1857714E-03	-3.1412698E-03	-3.3593696E-04	2.0405896E-04
	C_9	0.0000000E+00	0.0000000E+00	0.0000000E+00	0.0000000E+00
	C_{10}	0.0000000E+00	0.0000000E+00	0.0000000E+00	0.0000000E+00

Table 4.11.3 – Coefficients For Equation (4.11.5)

D_{pj}	Coefficients	Shell Thickness			
		19 mm (3/4 in.)	25 mm (1 in.)	50 mm (2 in.)	---
DN50 NPS 2	C_1	-4.0085121E+02	3.6782666E+00	1.0000000E+00	---
	C_2	3.5652906E+02	-1.2669560E+00	0.0000000E+00	---
	C_3	-1.3171601E+02	4.5491492E-01	0.0000000E+00	---
	C_4	2.6480374E+01	-6.2883969E-02	0.0000000E+00	---
	C_5	-3.1258388E+00	3.9401350E-03	0.0000000E+00	---
	C_6	2.1680455E-01	-9.3433360E-05	0.0000000E+00	---
	C_7	-8.1908188E-03	0.0000000E+00	0.0000000E+00	---
	C_8	1.3019970E-04	0.0000000E+00	0.0000000E+00	---
	C_9	0.0000000E+00	0.0000000E+00	0.0000000E+00	---
	C_{10}	0.0000000E+00	0.0000000E+00	0.0000000E+00	---
DN80 NPS 3	C_1	-1.5045135E+03	8.1206324E+00	-3.2789303E+03	---
	C_2	1.4487653E+03	-8.3943593E+00	3.4419302E+03	---
	C_3	-5.9846696E+02	3.7870074E+00	-1.5852932E+03	---
	C_4	1.3910417E+02	-7.0886182E-01	4.2063167E+02	---
	C_5	-1.9888205E+01	6.6972430E-02	-7.0855807E+01	---
	C_6	1.7922925E+00	-3.1488859E-03	7.8593168E+00	---
	C_7	-9.9521276E-02	5.8511141E-05	-5.7415834E-01	---
	C_8	3.1164737E-03	0.0000000E+00	2.6647325E-02	---
	C_9	-4.2181627E-05	0.0000000E+00	-7.1319265E-04	---
	C_{10}	0.0000000E+00	0.0000000E+00	8.3899940E-06	---
DN100 NPS 4	C_1	-3.5172282E+00	-2.5016604E+02	-5.3121462E+00	---
	C_2	4.3499616E+00	1.7178270E+02	3.4090615E+00	---
	C_3	-2.7157682E-01	-4.6844914E+01	-5.5605535E-01	---
	C_4	1.1186450E-02	6.6874346E+00	4.2156128E-02	---
	C_5	-7.1328067E-04	-5.2507555E-01	-1.2921987E-03	---
	C_6	2.2962890E-05	2.1526948E-02	6.6740230E-06	---
	C_7	0.0000000E+00	-3.6091550E-04	0.0000000E+00	---
	C_8	0.0000000E+00	0.0000000E+00	0.0000000E+00	---
	C_9	0.0000000E+00	0.0000000E+00	0.0000000E+00	---
	C_{10}	0.0000000E+00	0.0000000E+00	0.0000000E+00	---

4.11.9 Figures

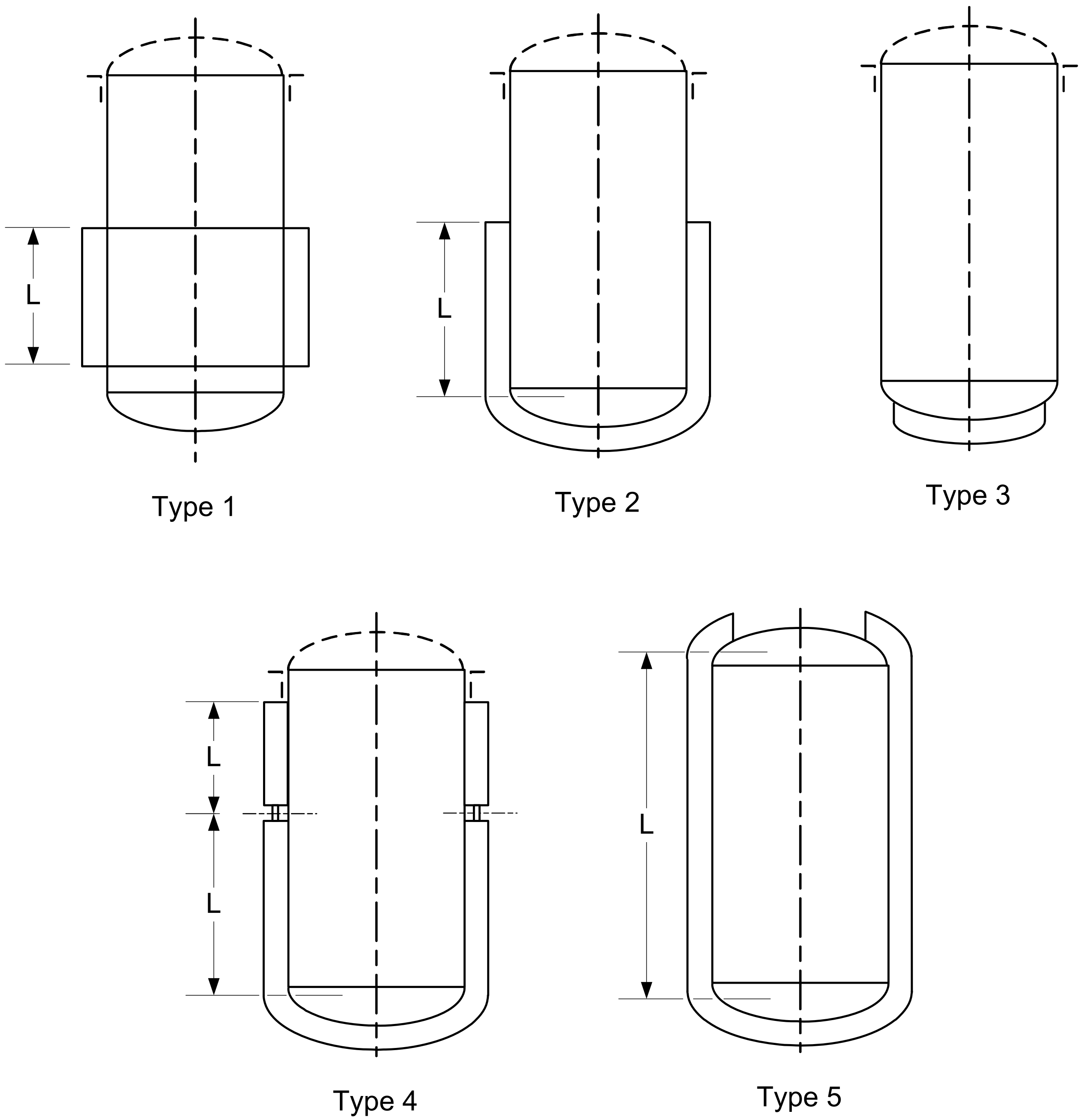
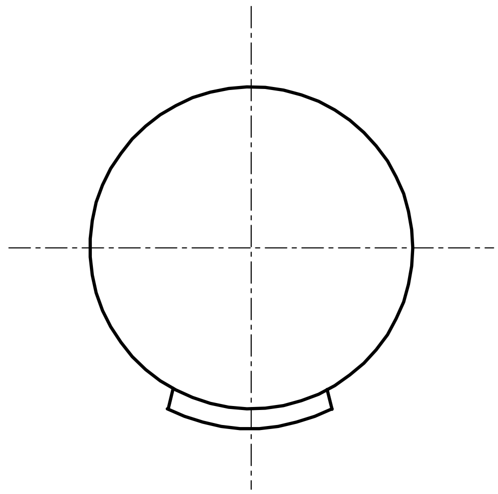
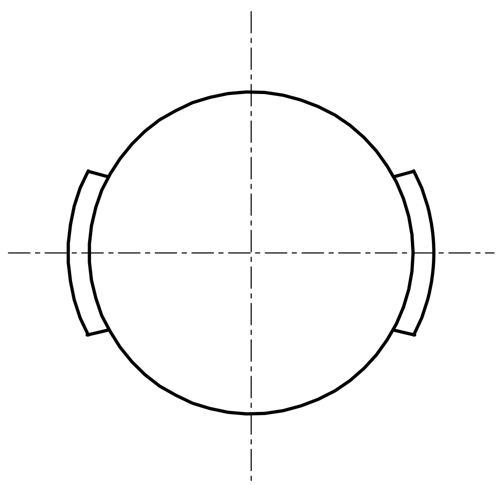


Figure 4.11.1 – Types of Jacketed Vessels



Continuous
Partial Jacket



Multiple or
Pod Type Jacket

Figure 4.11.2 – Types of Partial Jackets

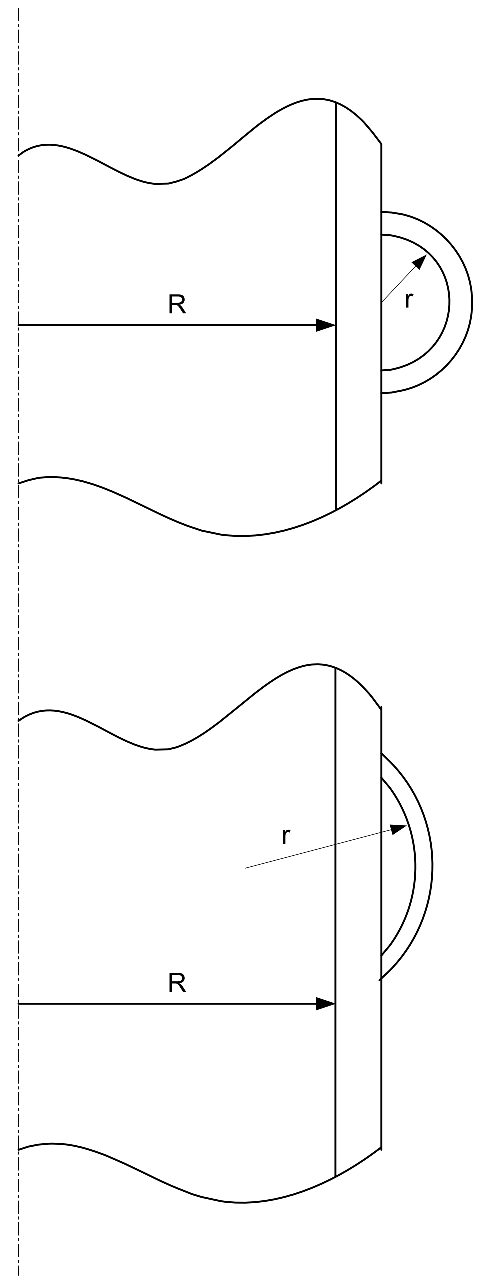


Figure 4.11.3 – Half Pipe Jackets

4.12 Design Rules for NonCircular Vessels

4.12.1 Scope

4.12.1.1 The procedures in paragraph 4.12 cover the design requirements for single wall vessels having a rectangular or obround cross section. The design rules cover the walls and parts of the vessels subject to pressure stresses including stiffening, reinforcing and staying members. All other types of loadings shall be evaluated in accordance with the design-by-analysis rules of Part 5.

4.12.1.2 The design rules in this paragraph cover noncircular vessels of the types shown in Table 4.12.1. Vessel configurations other than Types 1 to 12, illustrated in Figures 4.12.1 through 4.1.2.13, may be used. However, in this case, the design-by-analysis rules of Part 5 shall be used.

4.12.2 General Design Requirements

4.12.2.1 In the noncircular vessel configurations covered in this paragraph, the walls of the vessel can have different thicknesses. Therefore, the design of a noncircular vessel requires an iterative approach where the vessel configuration and wall thickness are initially set and the stresses at locations on the cross section are computed and compared to allowable values. If the allowable values are exceeded, the configuration and/or wall thickness are changed, and the stresses are revaluated. This process is continued until a final configuration including wall thickness is obtained where all allowable stress requirements are satisfied.

4.12.2.2 In the design rules of this paragraph, both membrane and bending stresses shall be computed at locations on the cross section. The membrane stress is added algebraically to the bending stress at both the outermost surface of the shell plate or reinforcement (when used) and the innermost surface of the shell plate to obtain two values of total stress. The total stresses at the section shall be compared to the allowable stress.

4.12.2.3 The total stresses (membrane plus bending) at the cross section of a vessel with and without reinforcement shall be calculated as follows.

a) For a vessel without reinforcement, the total stresses shall be determined at the inside and outside surfaces of the cross section of the shell plate.

b) For a vessel with reinforcement, when the reinforcing member has the same allowable stress as the vessel, the total stress shall be determined at the inside and outside surfaces of the composite cross section. The appropriate value of c (the location from the neutral axis) for the composite section properties shall be used in the bending equations. The total stresses at the inside and outside surfaces shall be compared to the allowable stress.

c) For a vessel with reinforcement, when the reinforcing member does not have the same allowable stress as the vessel, the total stresses shall be determined at the inside and outside surfaces of each component of the composite cross section. The appropriate value of c (the location from the neutral axis) for the composite section properties shall be used in the bending equations considering location of desired stress with respect to the composite section neutral axis. The total stresses at the inside and outside surfaces shall be compared to the allowable stress.

    1) For locations of stress below the neutral axis, the bending equation used to compute the stress shall be that considered acting on the inside surface.

    2) For locations of stress above the neutral axis, the bending equation used to compute the stress shall be that considered acting on the outside surface.

4.12.2.4 Particular attention shall be given to the effects of local internal and external loads and expansion differentials at design temperature, including reactions at supporting lugs, piping, and other types of attachments (see paragraph 4.12.1.1).

4.12.2.5 Except as otherwise specified in paragraph 4.12.8, vessel parts of noncircular cross section subject to external pressure shall be designed in accordance with Part 5.

4.12.2.6 The end closures for noncircular vessels covered in this paragraph shall be designed in accordance with the provisions of Part 5 except in cases where the ends are flat plates. For this case, the design rules of paragraph 4.6 shall be used except that 0.20 shall be used for the value of the C factor in all of the calculations.

4.12.2.7 The design equations in this paragraph are based on vessels in which the ratio of the long side to short side length is greater than four. These equations are conservatively applicable to vessels of aspect ratio less than four. Vessel side plates with aspect ratios less than four are strengthened by the interaction of the end closures and may be designed in accordance with the provisions of Part 5. Short unreinforced or unstayed vessels of rectangular cross section having an aspect ratio smaller than two may be designed in accordance with paragraph 4.12.5.

4.12.2.8 Bolted full side or end plates and flanges may be provided for vessels of rectangular cross section. Many acceptable configurations are possible. Therefore, rules for specific designs are not provided, and these parts shall be designed in accordance with Part 5. The analysis of the components shall consider thermal loads, gasket reactions, bolting forces, and resulting moments, as well as pressure and other mechanical loading.

4.12.2.9 Openings may be provided in vessels of noncircular cross section as follows:

a) Openings in noncircular vessels do not require reinforcement other than that inherent in the construction, provided they meet the conditions given in paragraph 4.5.2.

b) Compensation for openings in noncircular vessels must account for the bending strength as well as the membrane strength of the side with the opening. In addition, openings may significantly affect the stresses in adjacent sides. Because many acceptable configurations are possible, rules for specific designs are not provided and the design shall be in accordance with Part 5.

4.12.3 Requirements for Vessels With Reinforcement

4.12.3.1 Design rules are provided for Types 4, 5, and 6 (see Table 4.12.1) where the welded on reinforcement members are in a plane perpendicular to the long axis of the vessel; however, the spacing between reinforcing members need not be uniform. All reinforcement members attached to two opposite plates shall have the same moment of inertia. The design for any other type of reinforced rectangular cross section vessel shall be in accordance with Part 5 .

4.12.3.2 For a Type 4 vessel, when the side plate thicknesses are equal, the plates may be formed to a radius at the corners. The analysis is, however, carried out in the same manner as if the corners were not rounded. For corners that are cold formed, the provisions of Part 6 shall apply. For the special case where $L_1 = 0.0$, the analysis methodology for a Type 11 vessel shall be used.

4.12.3.3 A Type 5 vessel has rounded corners and non-continuous reinforcement. If continuous reinforcement is provided that follows the contour of the vessel, the design requirements for a Type 4 vessel shall be used.

4.12.3.4 For a Type 6 vessel, the corner region consists of a flat, chamfered segment joined to the adjacent sides by curved segments with constant radii. The chamfered segments shall be perpendicular to diagonal lines drawn through the points where the sides would intersect if they were extended.

4.12.3.5 Reinforcing members shall be placed on the outside of the vessel and shall be attached to the plates of the vessel by welding on each side of the reinforcing member. For continuous reinforcement, welding may be either continuous or intermittent. The total length of intermittent welding on each side of the reinforcing member shall be not less than one-half the length being reinforced on the shell. Welds on opposite sides of the reinforcing member may be either staggered or in-line and the distance between intermittent welds shall be no more than eight times the plate thickness of the plate being reinforced. For assuring the composite section properties, for non-continuous reinforcement, the welds must be capable of developing the necessary shear (see Manual of Steel Construction, AISC, American Institute of Steel Construction).

4.12.3.6 The maximum distance between reinforcing members is computed as follows.

a) The maximum distance between any reinforcing member centerlines is given by Equation (4.12.1). In the equations for calculating stresses for reinforced noncircular vessels, the value of p shall be taken as the sum of one-half the distances to the next reinforcing member on each side.

$$p = \min\left[p_1,\ p_2\right] \tag{4.12.1}$$

where

$$p_1 = t_1\sqrt{\frac{SJ_1}{P}} \qquad for\ H \geq p \tag{4.12.2}$$

$$p_1 = \frac{t_1}{\beta_1}\sqrt{\frac{SJ_1}{P}} \qquad for\ H < p \tag{4.12.3}$$

$$\beta_1 = \frac{H}{p_{b1}} \qquad for\ rectangular\ vessels \tag{4.12.4}$$

$$\beta_1 = \frac{2R}{p_{b1}} \qquad for\ obround\ vessels \tag{4.12.5}$$

$$p_{b1} = t_1\sqrt{\frac{2.1S}{P}} \tag{4.12.6}$$

$$J_1 = -0.26667 + \frac{24.222}{\left(\beta_{1\max}\right)} - \frac{99.478}{\left(\beta_{1\max}\right)^2} + \frac{194.59}{\left(\beta_{1\max}\right)^3} - \frac{169.99}{\left(\beta_{1\max}\right)^4} + \frac{55.822}{\left(\beta_{1\max}\right)^5} \tag{4.12.7}$$

$$\beta_{1\max} = \min\left[\max\left[\beta_1,\ \frac{1}{\beta_1}\right],\ 4.0\right] \tag{4.12.8}$$

$$p_2 = t_2\sqrt{\frac{SJ_2}{P}} \qquad for\ h \geq p \tag{4.12.9}$$

$$p_2 = \frac{t_2}{\beta_2}\sqrt{\frac{SJ_2}{P}} \qquad for\ h < p \tag{4.12.10}$$

$$\beta_2 = \frac{h}{p_{b2}} \qquad for\ rectangular\ vessels \tag{4.12.11}$$

$$\beta_2 = \frac{2L_2}{p_{b2}} \qquad for\ obround\ vessels \tag{4.12.12}$$

$$J_2 = -0.26667 + \frac{24.222}{\left(\beta_{2\max}\right)} - \frac{99.478}{\left(\beta_{2\max}\right)^2} + \frac{194.59}{\left(\beta_{2\max}\right)^3} - \frac{169.99}{\left(\beta_{2\max}\right)^4} + \frac{55.822}{\left(\beta_{2\max}\right)^5} \tag{4.12.13}$$

$$\beta_{2\max} = \min\left[\max\left[\beta_2,\ \frac{1}{\beta_2}\right],\ 4.0\right] \tag{4.12.14}$$

$$p_{b2} = t_2\sqrt{\frac{2.1S}{P}} \tag{4.12.15}$$

b) The allowable effective widths of the shell plate, w_1 and w_2, shall not be greater than the value given by Equation (4.12.16) or Equation (4.12.17) nor greater than the actual value of p if this value is less than that computed in paragraph 4.12.3.6.a. One-half of w shall be considered to be effective on each side of the reinforcing member centerline, but the effective widths shall not overlap. The effective width shall not be greater than the actual width available.

$$w_1 = \min\left[w_{\max},\ p_1\right] \tag{4.12.16}$$

$$w_2 = \min\left[w_{\max},\ p_2\right] \tag{4.12.17}$$

where

$$w_{\max} = \frac{t\Delta}{\sqrt{S_y}}\left(\frac{E_y}{E_{ya}}\right) \tag{4.12.18}$$

c) At locations, other than in the corner regions where the shell plate is in tension, the effective moments of inertia I_{11} and I_{21} of the composite section (reinforcement and shell plate acting together) shall be computed based on the values of w_1 and w_2 computed in paragraph 4.12.3.6.b. The equations given in paragraph 4.12.3.6.b do not include the effects of high-localized stresses. In the corner regions of some Type 4 configurations, the localized stresses may significantly exceed the calculated stress. Only a very small width of the shell plate may be effective in acting with the composite section in the corner region. The localized stresses in this region shall be evaluated using the principles of Part 5.

4.12.4 Requirements for Vessels With Stays

4.12.4.1 Three types of stayed construction are considered, Types 7, 8, and 11. In these types of construction the staying members may be plates welded to the side plates for the entire length of the vessel.

In this case, the stay plates shall not be constructed so as to create pressure-containing partitions. Alternatively, the stays may be bars of circular cross section fastened to the side plates on a uniform pitch designed in accordance with paragraph 4.9.

4.12.4.2 The Type 12 noncircular vessel is comprised of a cylindrical shell with a single stay plate that divides the cylinder into two compartments. The design rules ensure that the various vessel members will not be overstressed when there is full pressure in both vessel compartments or when there is full pressure in one compartment and zero pressure in the other compartment. Stresses may be computed only at the shell-plate junction since this is the location of maximum stress.

4.12.5 Requirements for Rectangular Vessels With Small Aspect Ratios

4.12.5.1 Type 1 and Type 2 noncircular vessels with aspect ratios of L_v/H or L_v/h between 1.0 and 2.0, and with flat heads welded to the sides may be designed using the procedure in paragraph 4.12.7 except that the following plate parameters shall be utilized in the calculations.

$$J_{2s} = J_2\left(x = \frac{L_v}{H}\right) \tag{4.12.19}$$

$$J_{2l} = J_2\left(x = \frac{L_v}{h}\right) \tag{4.12.20}$$

$$J_{3s} = J_3\left(x = \frac{L_v}{H}\right) \tag{4.12.21}$$

$$J_{3l} = J_3\left(x = \frac{L_v}{h}\right) \tag{4.12.22}$$

where

$$J_2(x) = -0.65979 + 1.0052x + 0.86072x^2 - 0.82362x^3 + 0.17483x^4 \tag{4.12.23}$$

$$J_3(x) = -0.37508 + 0.66706x + 0.99709x^2 - 0.84305x^3 + 0.17483x^4 \tag{4.12.24}$$

Note in the above nomenclature, $J_{2s} = J_2\left(x = \frac{L_v}{H}\right)$ is defined as computing J_{2s} using the function $J_2(x)$ evaluated at $x = \frac{L_v}{H}$.

4.12.5.2 For vessels with aspect ratios of L_v/H or L_v/h less than 1.0, the axis of the vessel shall be rotated so that the largest dimension becomes the length L_v, and the new ratios L_v/H or L_v/h are greater than or equal to 1.0. All stresses shall be recalculated using the new orientation.

4.12.6 Weld Joint Factors and Ligament Efficiency

4.12.6.1 The stress calculations for the noncircular vessel shall include a weld joint factor for weld locations and ligament efficiency for those locations containing holes. In the stress calculations two factors E_m and E_b are used to account for the weld joint factor and ligament efficiency that is to be applied to the

membrane and bending stresses, respectively. The weld joint factor shall be determined from paragraph 4.2 and the ligament efficiency shall be determined from paragraph 4.12.6.3. The correct combination of weld joint factor and ligament efficiencies to be used in the design is shown below.

a) If there is not a weld or a hole pattern at the stress calculation location, then:

$$E_m = 1.0 \tag{4.12.25}$$

$$E_b = 1.0 \tag{4.12.26}$$

b) If there is a weld, and there is not a hole pattern at the stress calculation location, then:

$$E_m = E \tag{4.12.27}$$

$$E_b = E \tag{4.12.28}$$

c) If there is not a weld, and there is a hole pattern at the stress calculation location, then:

$$E_m = e_m \tag{4.12.29}$$

$$E_b = e_b \tag{4.12.30}$$

d) If there is a weld and a hole pattern at the stress calculation location, then:

1) If e_m and e_b are less than the joint efficiency, E, which would be used if there were no ligaments in the plate, then use Equations (4.12.29) and(4.12.30).

2) If e_m and e_b are greater than the weld joint factor, E, which would be used if there were no ligaments in the plate, then use Equations (4.12.27) and(4.12.28).

4.12.6.2 Cases may arise where application of a weld joint factor, E, at non-welded locations results in unnecessarily increased plate thicknesses. If the butt weld occurs at one of the locations for which equations are provided in this paragraph, then no relief can be provided. However, if the weld occurs at some intermediate location, it is permissible to calculate the bending stress at the weld location and compare it to the allowable stress considering the weld joint factor in the calculation. An alternate location for computing stresses is provided for some of the noncircular geometry types, and is identified as “Maximum Membrane And Bending Stresses – Defined Locations” in the stress calculation tables. The value of X or Y to be used in the equations is the distance from the midpoint of the side to the location of the weld joint.

4.12.6.3 The ligament efficiency factors e_m and e_b, for membrane and bending stresses, respectively, shall only be applied to the calculated stresses for the plates containing the ligaments.

a) For the case of uniform diameter holes, the ligament efficiency factors e_m and e_b shall be the same and computed in accordance with paragraph 4.10.

b) For the case of multi-diameter holes, the neutral axis of the ligament may no longer be at mid-thickness of the plate; the bending stress is higher at one of the plate surfaces than at the other surface. Therefore, for multi-diameter holes, the ligament efficiency factor shall be computed using the following equations.

1) The ligament efficiency of plate with multi-diameter holes subject to membrane stress is computed as follows.

$$e_m = \frac{\left(p_h - D_E\right)}{p_h} \tag{4.12.31}$$

where

$$D_E = \frac{1}{t}\left(d_0 T_0 + d_1 T_1 + d_2 T_2 + \ldots.. + d_n T_n\right) \tag{4.12.32}$$

2) The ligament efficiency and location from the neutral axis of a plate with multi-diameter holes (see Figure 4.12.14) subject to bending stress is computed as follows.

$$e_b = \frac{\left(p_h - D_E\right)}{p_h} \tag{4.12.33}$$

where

$$D_E = p_h - \frac{6I_E}{t^2 c_E} \tag{4.12.34}$$

$$\begin{aligned} I_E = &\frac{1}{12}\left(b_0 T_0^3 + b_1 T_1^3 + b_2 T_2^3 + \ldots.. + b_n T_n^3\right) + \\ &b_0 T_0 \left(\frac{T_0}{2} + T_1 + T_2 + \ldots.. + T_n - \overline{X}\right)^2 + \ b_1 T_1 \left(\frac{T_1}{2} + T_2 + \ldots.. + T_n - \overline{X}\right)^2 + \\ &b_2 T_2 \left(\frac{T_2}{2} + \ldots.. + T_n - \overline{X}\right)^2 + \ b_n T_n \left(\overline{X} - \frac{T_n}{2}\right)^2 \end{aligned} \tag{4.12.35}$$

$$\overline{X} = \begin{bmatrix} b_0 T_0 \left(\frac{T_0}{2} + T_1 + T_2 + \ldots.. + T_n\right) + \\ b_1 T_1 \left(\frac{T_1}{2} + T_2 + \ldots.. + T_n\right) + \\ b_2 T_2 \left(\frac{T_2}{2} + \ldots.. + T_n\right) + b_n T_n \left(\frac{T_n}{2}\right) \end{bmatrix} \cdot \left[b_0 T_0 + b_1 T_1 + b_2 T_2 + \ldots.. + b_n T_n\right]^{-1} \tag{4.12.36}$$

where

$$b_0 = p_h - d_0 \tag{4.12.37}$$

$$b_1 = p_h - d_1 \tag{4.12.38}$$

$$b_2 = p_h - d_2 \tag{4.12.39}$$

$$b_n = p_h - d_n \tag{4.12.40}$$

$$c_E = \max\left[\bar{X},\ \left(t-\bar{X}\right)\right] \tag{4.12.41}$$

If T_o is measured from the inside surface, then

$$c_i = \bar{X} \tag{4.12.42}$$

$$c_o = t - \bar{X} \tag{4.12.43}$$

If T_o is measured from the outside surface, then

$$c_i = t - \bar{X} \tag{4.12.44}$$

$$c_o = \bar{X} \tag{4.12.45}$$

c) Rows of holes may be located in regions of relatively low bending moments to keep the required plate thickness to a minimum. Therefore, it is permissible to calculate the stresses at the centerline of each row of holes closest to the locations where the highest bending moments occurs (i.e. at the midpoint of the sides and at the corners). If the diameter of all the holes is not the same, the stresses must be calculated for each set of e_m and e_b values.

d) The applied gross area stresses may be calculated using the same procedure as for calculating the stresses at a weld joint (see paragraph 4.12.3.2). The value of X or Y to be used in the equations is the distance from the midpoint of the side to the plane containing the centerlines of the holes.

4.12.7 Design Procedure

4.12.7.1 A procedure that can be used to design a noncircular vessel subject to internal pressure is shown below.

a) STEP 1 – Determine the design pressure and temperature.

b) STEP 2 – Determine the configuration of the noncircular vessel by choosing a Type from Table 4.12.1.

c) STEP 3 – Determine the initial configuration (i.e. width, height, length, etc.) and wall thicknesses of the pressure containing plates.

   1) If the vessel has stiffeners, then determine the spacing (see paragraph 4.12.3) and size of the stiffeners.

   2) If the vessel has stays, then determine the stay type and configuration (see paragraph 4.12.4), and check the stay plate welds using paragraph 4.2.

   3) If the vessel aspect ratio is less than two, then determine the plate parameters in paragraph 4.12.5.

d) STEP 4 – Determine the location of the neutral axis from the inside and outside surfaces.

   1) If the section under evaluation has stiffeners, then c_i and c_o are determined from the cross section of the combined plate and stiffener section using strength of materials concepts.

   2) If the section under evaluation has multi-diameter holes, then c_i and c_o are determined from paragraph 4.12.6.3.

   3) If the section under evaluation does not have a stiffener, does not have holes, or has uniform diameter holes, then $c_i = c_o = t/2$ where t is the thickness of the plate.

e) STEP 5 – Determine the weld joint factor and ligaments efficiencies, as applicable (see paragraph 4.12.6), and determine the factors E_m or E_b.

f) STEP 6 – Complete the stress calculation for the selected noncircular vessel Type (see Table 4.12.1), and check the acceptance criteria. If the criteria are satisfied, then the design is complete. If the criteria are not satisfied, then modify the plate thickness and/or stiffener size and go to STEP 3 and repeat the calculation. Continue this process until a design is achieved that satisfies the acceptance criteria.

4.12.7.2 If the vessel is subject to external pressure, the additional requirements of paragraph 4.12.8 shall be satisfied.

4.12.8 Noncircular Vessels Subject to External Pressure

4.12.8.1 Rectangular vessel Types 1 and 2 subject to external pressure shall meet the following requirements.

a) The stresses shall be calculated in accordance with Tables 4.12.2 and 4.12.3 except that the design external pressure shall be substituted for P. These computed stresses shall meet the acceptance criteria defined in these tables.

b) The four side plates and the two end plates shall be checked for stability in accordance with Equation (4.12.46). The required calculations for S_{mA}, S_{mB}, S^*_{crA}, S^{**}_{crA}, S^*_{crB} and S^{**}_{crB} are shown in Table 4.12.15. In the equations, the subscript A is used to identify stress or load acting in a direction parallel to the long dimension of the panel being considered and the subscript B is used to identify stress or load acting in a direction parallel to the short dimension of the panel being considered. In the calculations, the plate thickness t shall be adjusted if the plate is perforated. This can be accomplished by multiplying t by e_m in the equations for S_{mA} and S_{mB}. It is not necessary to make this adjustment in the equations for S_{crA} and S_{crB}.

$$\frac{2S_{mA}}{S_{crA}}+\frac{2S_{mB}}{S_{crB}}\le 1.0 \tag{4.12.46}$$

where

$$S_{crA}=S^*_{crA} \qquad when\ S^*_{crA}\le\frac{S_y}{2} \tag{4.12.47}$$

$$S_{crA}=S^{**}_{crA} \qquad when\ S^*_{crA}>\frac{S_y}{2} \tag{4.12.48}$$

$$S_{crB}=S^*_{crB} \qquad when\ S^*_{crB}\le\frac{S_y}{2} \tag{4.12.49}$$

$$S_{crB}=S^{**}_{crB} \qquad when\ S^*_{crB}>\frac{S_y}{2} \tag{4.12.50}$$

c) In addition to checking each of the four side plates and the two end plates for stability, the cross section shall be checked for column stability using the following equations. Equation (4.12.52) applies to vessels where the long plate thicknesses are equal. If the thicknesses are not equal, replace $2t_2$ with $\left(t_2+t_{22}\right)$ in the equation.

$$\frac{S_a}{F_a}+\frac{S_b}{S\left(1-\frac{S_a}{F_e^*}\right)}\leq 1.0 \tag{4.12.51}$$

where

$$S_a=\frac{P_e\left(h+2t_1\right)\left(H+2t_2\right)}{2t_1\left(H+2t_2\right)+2t_2\left(h+2t_1\right)} \tag{4.12.52}$$

$$S_b=\frac{\left[P_e\left(h+2t_1\right)\left(H+2t_2\right)\bar{y}\right]c_e}{I_e} \tag{4.12.53}$$

$$C_c=\sqrt{\frac{2\pi^2 E_y}{S_y}} \tag{4.12.54}$$

$$F_a=\frac{\left[1-\frac{1}{2C_c^2}\left(\frac{2L_v}{R_{ge}}\right)^2\right]S_y}{\frac{5}{3}+\frac{3}{8C_c}\left(\frac{2L_v}{R_{ge}}\right)-\frac{1}{8C_c^3}\left(\frac{2L_v}{R_{ge}}\right)^3} \qquad when\ \frac{2L_v}{R_{ge}}\leq C_c \tag{4.12.55}$$

$$F_a=\frac{12\pi^2 E_y}{23\left(\frac{2L_v}{R_{ge}}\right)^2} \qquad when\ \frac{2L_v}{R_{ge}}>C_c \tag{4.12.56}$$

$$F_e^*=\frac{12\pi^2 E_y}{23\left(\frac{2L_v}{R_{ge}}\right)^2} \tag{4.12.57}$$

4.12.9 Rectangular Vessels With Two or More Compartments of Unequal Size

Typical rectangular cross section vessels having unequal compartments are shown in Figure 4.12.15. These types of vessels shall be qualified using either of the two methods shown below.

a) A design can be qualified by selecting the compartment having the maximum dimensions and analyzing the vessel as a Type 7 for the case of a two-compartment vessel or Type 8 for the case of a vessel with more than two compartments. For example, if the vessel has two unequal compartments, use the geometry for a Type 7 with each compartment having the maximum dimension of the actual vessel. For a vessel with more than two compartments, use the geometry for a Type 8 with three compartments having the maximum dimensions of the actual vessel. Thus a five or six compartment vessel would be analyzed as if it had only three compartments.

b) The vessel can be designed in accordance with Part 5 .

4.12.10 Fabrication

4.12.10.1 Provided the requirements of the applicable Parts of this Division are satisfied, fabrication methods other than welding are permitted.

4.12.10.2 Category A joints may be of Type 3 when the thickness does not exceed 16 mm (0.625 in.).

4.12.11 Nomenclature

4.12.11.1 The nomenclature used in this paragraph is defined below except for computed stresses. The nomenclature for computed stress is defined in paragraph 4.12.11.2.

A_1 cross-sectional area of the reinforcing member associated with t_1.

A_2 cross-sectional area of the reinforcing member associated with t_2.

b unit width per cross section. In the equations for the areas, moments of inertia, and bending moments for all vessel configurations without external reinforcements are given for cross sections with a unit width.

C stress factor for braced and stayed surfaces (see Table 4.9.1).

c_e location from the neutral axis to the outer most surface of a composite section associated t_v, R, A_R.

c_i distance from the neutral axis to the inside surface of the shell or reinforcing member on the short side, long side, curved element, or stay plate as applicable (e.g. for a plate with uniform holes without a stiffener, $c = t/2$ where t is the thickness of the plate); the sign of this parameter is always positive (the sign for the bending stress is included in the applicable equation).

c_o distance from the neutral axis to the outside surface of the shell or reinforcing member on the short side, long side, curved element, or stay plate as applicable (e.g. for a plate with uniform holes without a stiffener, $c = t/2$ where t is the thickness of the plate); the sign of this parameter is always positive (the sign for the bending stress is included in the applicable equation).

Δ effective width coefficient (see Table 4.12.14)

d_j hole diameter jth location.

e_b bending stress ligament efficiency of a hole pattern.

e_m membrane stress ligament efficiency of a hole pattern.

E weld joint factor.

E_b factor applied to the bending stress to account for a ligament or weld joint factor.

E_m factor applied to the membrane stress to account for a ligament or weld joint factor.

E_y Young's Modulus from Annex 3.E at design temperature.

E_{ya} Young's Modulus from Annex 3.E at ambient temperature.

H inside length of the short side of a rectangular vessel. For Types 5 and 6, $H = 2(L_1 + L_{11})$ and for Type 10 $H = 2R$.

H_1 centroidal length of the reinforcing member on the short side of a rectangular vessel.

h inside length of the long side of a rectangular vessel. For Types 5 and 6, $h = 2(L_2 + L_{21})$ and for Type 10 $H = 2L_2$.

h_1 centroidal length of the reinforcing member on the long side of a rectangular vessel.

I_e least moment of inertia of noncircular cross-section vessel.

I_1 moment of inertia of strip thickness t_1.

I_2 moment of inertia of strip thickness t_2.

I_{22} moment of inertia of strip thickness t_{22}.

I_3 moment of inertia of strip thickness t_3.

I_{11} moment of inertia of combined reinforcing member and effective with of plate w of thickness t_1.

I_{21} moment of inertia of combined reinforcing member and effective with of plate w of thickness t_2.

L_1 half-length of the short side of a rounded vessel without reinforcement or the half-length of reinforcement on the short side for a reinforced rectangular vessel.

L_2 half-length of the long side of a rounded vessel without reinforcement or the half-length of reinforcement on the long side for a reinforced rectangular vessel.

L_3 half-length dimension of the short side of Type 5 and Type 6 rectangular vessel.

L_4 half-length dimension of the long side of Type 5 and Type 6 rectangular vessel.

L_{11} length measured from the edge of the reinforcement to the end of the straight side of the short side of a Type 5 and Type 6 rectangular vessel.

L_{21} length measured from the edge of the reinforcement to the end of the straight side of the long side of a Type 5 and Type 6 rectangular vessel.

L_v length of the vessel.

M_A bending moment at the mid-side of the long side, a positive sign indicates a compressive stress on the outside surface of the plate.

N rectangular vessel parameter.

P internal design pressure.

P_1 internal design pressure of a two compartment vessel where $P_1 \geq P_2$.

P_2 internal design pressure of a two compartment vessel where $P_1 \geq P_2$.

p distance between reinforcing members; plate width between edges of reinforcing members

p_h pitch distance between holes.

R inside radius.

R_{ge} least radius of gyration of a noncircular cross-section vessel.

r radius to centroidal axis of reinforcement member on obround vessel.

S allowable stress from Annex 3.A at the design temperature.

S_y yield stress at the design temperature evaluated in accordance Annex 3.D .

t plate thickness.

t_1 thickness of the short side plate.

t_2 thickness of the long side plate.

t_3 thickness or diameter of staying member.

t_4 thickness or diameter of staying member.

t_5 thickness of end closure plate or head of vessel.

t_{22} thickness of the thicker long side plate.

T_j hole depth jth location.

ν Poisson's ratio.

w width of plate included in the moment of inertia calculation of the reinforced section.

$\overline{y}$ distance from geometric center of end plate to centroid of cross-sectional area of a rectangular vessel.

4.12.11.2 The nomenclature for all computed stress quantities is shown in the following tables and figures.

a) For Types 1, 4, 7, and 8 noncircular vessels see Tables 4.12.2, 4.12.5, 4.12.8, and 4.12.9 and Figures 4.12.1, 4.12.4, 4.12.8, and 4.12.9

b) For the Type 2 noncircular vessel, see Table 4.12.3 and Figure 4.12.2

c) For the Type 3 noncircular vessel, see Table 4.12.4 and Figure 4.12.3

d) For the Type 5 noncircular vessel, see Table 4.12.6 and Figure 4.12.5

e) For the Type 6 noncircular vessel, see Table 4.12.7 and Figures 4.12.6 and 4.12.)

f) For the Types 9, 10, and 11 noncircular vessels, see Tables 4.12.10, 4.12.11. and 4.12.12 and Figures 4.12.10, 4.12.11, and 4.12.12

g) For the Type 12 noncircular vessels, see Table 4.12.13 and Figure 4.12.13

h) For the compressive stress calculations for Type 1 and 2 see Table 4.12.15

4.12.12 Tables

Table 4.12.1 – Noncircular Vessel Configurations And Types

Configuration	Type	Figure Number	Table Containing Design Rules
Rectangular cross-section in which the opposite sides have the same wall thickness. Two opposite sides may have a wall thickness different than that of the other two opposite sides.	1	4.12.1	4.12.2
Rectangular cross-section in which two opposite members have the same thickness and the other two members have two different thicknesses.	2	4.12.2	4.12.3
Rectangular cross section having uniform wall thickness and corners bent to a radius. For corners which are cold formed, the provisions Part 6 shall apply	3	4.12.3	4.12.4
Rectangular cross-section similar to Type 1 but reinforced by stiffeners welded to the sides.	4	4.12.4	4.12.5
Rectangular cross-section similar to Type 3 but externally reinforced by stiffeners welded to the flat surfaces of the vessel.	5	4.12.5	4.12.6
Rectangular cross section with chamfered corner segments (octagonal cross-section) joined to the adjacent sides by small curved segments with constant radii and reinforced by stiffeners welded to the flat surfaces of the vessel.	6	4.12.6, 4.12.7	4.12.7
Rectangular cross section similar to Type 1 but having two opposite sides stayed at mid-length.	7	4.12.8	4.12.8
Rectangular cross section similar to Type 1 but having two opposite sides stayed at the third points.	8	4.12.9	4.12.9
Obround cross-section in which the opposite sides have the same wall thickness. The flat sidewalls may have a different thickness than the semi-cylindrical parts.	9	4.12.10	4.12.10
Obround cross-section similar to Type 9 but reinforced by stiffeners welded to the curved and flat surfaces of the vessel.	10	4.12.11	4.12.11
Obround cross-section similar to Type 9 but having the flat side plates stayed at mid-length.	11	4.12.12	4.12.12
Circular Section With A Single Stay Plate	12	4.12.13	4.12.13

Table 4.12.2 – Stress Calculations And Acceptance Criteria For Type 1 Noncircular Vessels (Rectangular Cross Section)

Membrane And Bending Stresses – Critical Locations of Maximum Stress
$S_m^s = \dfrac{Ph}{2t_1 E_m}$ $S_{bi}^{sC} = -S_{bo}^{sC}\left(\dfrac{c_i}{c_o}\right) = \dfrac{PbJ_{2s}c_i}{12I_1E_b}\left[-1.5H^2 + h^2\left(\dfrac{1+\alpha^2K}{1+K}\right)\right]$ $S_{bi}^{sB} = -S_{bo}^{sB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbh^2J_{3s}c_i}{12I_1E_b}\left[\dfrac{1+\alpha^2K}{1+K}\right]$ $S_m^l = \dfrac{PH}{2t_2 E_m}$ $S_{bi}^{lA} = -S_{bo}^{lA}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbh^2J_{2l}c_i}{12I_2E_b}\left[-1.5 + \left(\dfrac{1+\alpha^2K}{1+K}\right)\right]$ $S_{bi}^{lB} = -S_{bo}^{lB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbh^2J_{3l}c_i}{12I_2E_b}\left[\dfrac{1+\alpha^2K}{1+K}\right]$
Membrane And Bending Stresses – Defined Locations for Stress Calculation
$S_{bi}^{sX} = -S_{bo}^{sX}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbc_i}{12I_1E_b}\left[-1.5H^2 + h^2\left(\dfrac{1+\alpha^2K}{1+K}\right) + 6X^2\right]$ $S_{bi}^{lY} = -S_{bo}^{lY}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbc_i}{12I_2E_b}\left[-1.5h^2 + h^2\left(\dfrac{1+\alpha^2K}{1+K}\right) + 6Y^2\right]$
Equation Constants
$I_1 = \dfrac{bt_1^3}{12}$ $J_{2s} = 1.0$ $(see\ paragraph\ 4.12.5\ for\ exception)$ $I_2 = \dfrac{bt_2^3}{12}$ $J_{3s} = 1.0$ $(see\ paragraph\ 4.12.5\ for\ exception)$ $K = \dfrac{I_1}{I_2}\alpha$ $J_{2l} = 1.0$ $(see\ paragraph\ 4.12.5\ for\ exception)$ $\alpha = \dfrac{H}{h}$ $J_{3l} = 1.0$ $(see\ paragraph\ 4.12.5\ for\ exception)$

Table 4.12.2 – Stress Calculations And Acceptance Criteria For Type 1 Noncircular Vessels (Rectangular Cross Section)

Acceptance Criteria – Critical Locations of Maximum Stress	
$S_m^s \le S$	$S_m^l \le S$
$S_m^s + S_{bi}^{sC} \le 1.5S$	$S_m^l + S_{bi}^{lA} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$	$S_m^l + S_{bo}^{lA} \le 1.5S$
$S_m^s + S_{bi}^{sB} \le 1.5S$	$S_m^l + S_{bi}^{lB} \le 1.5S$
$S_m^s + S_{bo}^{sB} \le 1.5S$	$S_m^l + S_{bo}^{lB} \le 1.5S$
Acceptance Criteria – Defined Locations for Stress Calculation	
$S_m^s + S_{bi}^{sX} \le 1.5S$	$S_m^l + S_{bi}^{lY} \le 1.5S$
$S_m^s + S_{bo}^{sX} \le 1.5S$	$S_m^l + S_{bo}^{lY} \le 1.5S$

Nomenclature For Stress Results	
S_m^s	membrane stress in the short side.
S_{bi}^{sB}, S_{bo}^{sB}	bending stress in the short side at point B on the inside and outside surfaces, respectively.
S_{bi}^{sC}, S_{bo}^{sC}	bending stress in the short side at point C on the inside and outside surfaces, respectively.
S_{bi}^{sX}, S_{bo}^{sX}	bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.
S_m^l	membrane stress in the long side.
S_{bi}^{lB}, S_{bo}^{lB}	bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lA}, S_{bo}^{lA}	bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY}	bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^{st}	membrane stress in the stay bar or plate, as applicable.

Table 4.12.3 – Stress Calculations And Acceptance Criteria For Type 2 Noncircular Vessels (Rectangular Cross Section With Unequal Side Plate Thicknesses)

Membrane And Bending Stresses – Critical Locations of Maximum Stress

$$S_m^s = \frac{Ph}{2t_1 E_m}$$

$$S_{bi}^{sB} = -S_{bo}^{sB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{3s} c_i}{4NI_1 E_b}\left[\left(K_2 - k_1 k_2\right) + \alpha^2 k_2\left(K_2 - k_2\right)\right]$$

$$S_{bi}^{sC} = -S_{bo}^{sC}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{3s} c_i}{4NI_1 E_b}\left[\left(K_1 k_1 - k_2\right) + \alpha^2 k_2\left(K_1 - k_2\right)\right]$$

$$S_m^{l2} = \frac{P}{8NHt_2 E_m}\left\{4NH^2 - 2h^2\left[\left(K_2 + k_2\right) - k_1\left(K_1 + k_2\right) + \alpha^2 k_2\left(K_2 - K_1\right)\right]\right\}$$

$$S_{bi}^{lD} = -S_{bo}^{lD}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{2l} c_i}{8NI_2 E_b}\left\{2\left[\left(K_1 k_1 - k_2\right) + \alpha^2 k_2\left(K_1 - k_2\right)\right] - N\right\}$$

$$S_{bi}^{lC} = -S_{bo}^{lC}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{3l} c_i}{4NI_2 E_b}\left[\left(K_1 k_1 - k_2\right) + \alpha^2 k_2\left(K_1 - k_2\right)\right]$$

$$S_m^{l22} = \frac{P}{8NHt_{22} E_m}\left\{4NH^2 - 2h^2\left[-\left(K_2 + k_2\right) + k_1\left(K_1 + k_2\right) - \alpha^2 k_2\left(K_2 - K_1\right)\right]\right\}$$

$$S_{bi}^{lA} = -S_{bo}^{lA}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{2l} c_i}{8NI_{22} E_b}\left\{2\left[\left(K_2 - k_1 k_2\right) + \alpha^2 k_2\left(K_2 - k_2\right)\right] - N\right\}$$

$$S_{bi}^{lB} = -S_{bo}^{lB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2 J_{3l} c_i}{4NI_{22} E_b}\left[\left(K_2 - k_1 k_2\right) + \alpha^2 k_2\left(K_2 - k_2\right)\right]$$

Membrane And Bending Stresses – Defined Locations for Stress Calculation

$$S_{bi}^{lY_2} = -S_{bo}^{lY_2}\left(\frac{c_i}{c_o}\right) = \frac{Pbc_i}{2I_2 E_b}\left\{\frac{h^2}{2N}\left[\left(K_1 k_1 - k_2\right) + \alpha^2 k_2\left(K_1 - k_2\right)\right] - \frac{h^2}{4} + Y_2^2\right\}$$

$$S_{bi}^{lY_{22}} = -S_{bi}^{lY_{22}}\left(\frac{c_i}{c_o}\right) = \frac{Pbc_i}{2I_{22} E_b}\left\{h^2\left[\left(K_2 - k_1 k_2\right) + \alpha^2 k_2\left(K_2 - k_2\right)\right] - \frac{h^2}{4} + Y_{22}^2\right\}$$

Table 4.12.3 – Stress Calculations And Acceptance Criteria For Type 2 Noncircular Vessels (Rectangular Cross Section With Unequal Side Plate Thicknesses)

Equation Constants		
$I_1 = \dfrac{bt_1^3}{12}$ $I_{22} = \dfrac{bt_{22}^3}{12}$ $k_1 = \dfrac{I_{22}}{I_2}$ $k_2 = \dfrac{I_{22}\alpha}{I_1}$	$K_1 = 2k_2 + 3$ $K_2 = 3k_1 + 2k_2$ $N = K_1 K_2 - k_2^2$ $J_{2s} = 1.0 \ \ (see\ paragraph\ 4.12.5\ for\ exception)$ $J_{3s} = 1.0 \ \ (see\ paragraph\ 4.12.5\ for\ exception)$ $J_{2l} = 1.0 \ \ (see\ paragraph\ 4.12.5\ for\ exception)$ $J_{3l} = 1.0 \ \ (see\ paragraph\ 4.12.5\ for\ exception)$	
Acceptance Criteria – Critical Locations of Maximum Stress		
$S_m^s \le S$	$S_m^{l2} \le S$	$S_m^{l22} \le S$
$S_m^s + S_{bi}^{sB} \le 1.5S$	$S_m^{l2} + S_{bi}^{lC} \le 1.5S$	$S_m^{l22} + S_{bi}^{lA} \le 1.5S$
$S_m^s + S_{bo}^{sB} \le 1.5S$	$S_m^{l2} + S_{bo}^{lC} \le 1.5S$	$S_m^{l22} + S_{bo}^{lA} \le 1.5S$
$S_m^s + S_{bi}^{sC} \le 1.5S$	$S_m^{l2} + S_{bi}^{lD} \le 1.5S$	$S_m^{l22} + S_{bi}^{lB} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$	$S_m^{l2} + S_{bo}^{lD} \le 1.5S$	$S_m^{l22} + S_{bo}^{lB} \le 1.5S$
Acceptance Criteria – Defined Locations for Stress Calculation		
Not Applicable	$S_m^{l2} + S_{bi}^{lY_2} \le 1.5S$ $S_m^{l2} + S_{bo}^{lY_2} \le 1.5S$	$S_m^{l22} + S_{bi}^{lY_{22}} \le 1.5S$ $S_m^{l22} + S_{bo}^{lY_{22}} \le 1.5S$

Table 4.12.3 – Stress Calculations And Acceptance Criteria For Type 2 Noncircular Vessels (Rectangular Cross Section With Unequal Side Plate Thicknesses)

Nomenclature For Stress Results	
S_m^s	membrane stress in the short side.
S_{bi}^{sB} , S_{bo}^{sB}	bending stress in the short side at point B on the inside and outside surfaces, respectively.
S_{bi}^{sC} , S_{bo}^{sC}	bending stress in the short side at point C on the inside and outside surfaces, respectively.
S_m^{l2}	membrane stress in the long side with thickness t_2 .
S_{bi}^{lD} , S_{bo}^{lD}	bending stress in the long side at point D on the inside and outside surfaces, respectively.
S_{bi}^{lC} , S_{bo}^{lC}	bending stress in the long side at point C on the inside and outside surfaces, respectively.
$S_{bi}^{lY_2}$, $S_{bo}^{lY_2}$	bending stress in the long side at a point defined by Y_2 on the inside and outside surfaces, respectively.
S_m^{l22}	membrane stress in the long side with thickness t_{22} .
S_{bi}^{lD} , S_{bo}^{lD}	bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lC} , S_{bo}^{lC}	bending stress in the long side at point B on the inside and outside surfaces, respectively.
$S_{bi}^{lY_{22}}$, $S_{bo}^{lY_{22}}$	bending stress in the long side at a point defined by Y_{22} on the inside and outside surfaces, respectively.

Table 4.12.4 – Stress Calculations And Acceptance Criteria For Type 3 Noncircular Vessels (Chamfered Rectangular Cross Section)

Membrane And Bending Stresses – Critical Locations of Maximum Stress
$S_m^s = \dfrac{P(R+L_2)}{t_1 E_m}$
$S_{bi}^{sC} = -S_{bo}^{sC}\left(\dfrac{c_i}{c_o}\right) = \dfrac{bc_i}{2I_1E_b}\left[2M_A + P\left(2RL_2 - 2RL_1 + L_2^2\right)\right]$
$S_{bi}^{sD} = -S_{bo}^{sD}\left(\dfrac{c_i}{c_o}\right) = \dfrac{bc_i}{2I_1E_b}\left[2M_A + P\left(L_2^2 + 2RL_2 - 2RL_1 - L_1^2\right)\right]$
$S_m^l = \dfrac{P(R+L_1)}{t_1 E_m}$
$S_{bi}^{lA} = -S_{bo}^{lA}\left(\dfrac{c_i}{c_o}\right) = \dfrac{bM_Ac_i}{I_1E_b}$
$S_{bi}^{lB} = -S_{bo}^{lB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{bc_i}{2I_1E_b}\left[2M_A + PL_2^2\right]$
$S_m^c = \dfrac{P}{t_1 E_m}\left(R + \sqrt{L_2^2 + L_1^2}\right)$
$S_{bi}^{cBC} = -S_{bo}^{cBC}\left(\dfrac{c_i}{c_o}\right) = \dfrac{bM_rc_i}{I_1E_b}$
Membrane And Bending Stresses – Defined Locations for Stress Calculation
$S_{bi}^{sX} = -S_{bi}^{sX}\left(\dfrac{c_i}{c_o}\right) = \dfrac{bc_i}{I_1E_b}\left[M_A + \dfrac{P}{2}\left(L_2^2 + 2RL_2 - 2RL_1 - L_1^2 + X^2\right)\right]$
$S_{bi}^{lY} = -S_{bi}^{lY}\left(\dfrac{c_i}{c_o}\right) = \dfrac{bc_i}{I_1E_b}\left[M_A + \dfrac{PY^2}{2}\right]$

Table 4.12.4 – Stress Calculations And Acceptance Criteria For Type 3 Noncircular Vessels (Chamfered Rectangular Cross Section)

Equation Constants

$$I_1 = \frac{bt_1^3}{12}$$

$$M_A = \frac{-PL_1^2\left(6\phi^2\alpha_3 - 3\pi\phi^2 + 6\phi^2 + \alpha_3^3 + 3\alpha_3^2 - 6\phi - 2 + 1.5\pi\phi\alpha_3^2 + 6\phi\alpha_3\right)}{3\left(2\alpha_3 + \pi\phi + 2\right)}$$

$$\phi = \frac{R}{L_1}$$

$$\alpha_3 = \frac{L_2}{L_1}$$

$$M_r = M_A + P\left(R\left\{L_2\cos\theta - L_1\left(1-\sin\theta\right)\right\} + \frac{L_2^2}{2}\right)$$

$$M_r \ is \ a \ maximum \ at \ \theta = \arctan\left(\frac{L_1}{L_2}\right)$$

Acceptance Criteria – Critical Locations of Maximum Stress

$S_m^s \le S$	$S_m^l \le S$	$S_m^c \le S$
$S_m^s + S_{bi}^{sC} \le 1.5S$	$S_m^l + S_{bi}^{lA} \le 1.5S$	$S_m^c + S_{bi}^{cBC} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$	$S_m^l + S_{bo}^{lA} \le 1.5S$	$S_m^c + S_{bo}^{cBC} \le 1.5S$
$S_m^s + S_{bi}^{sD} \le 1.5S$	$S_m^l + S_{bi}^{lB} \le 1.5S$	
$S_m^s + S_{bo}^{sD} \le 1.5S$	$S_m^l + S_{bo}^{lB} \le 1.5S$	

Acceptance Criteria – Defined Locations for Stress Calculation

$S_m^s + S_{bi}^{sX} \le 1.5S$	$S_m^l + S_{bi}^{lX} \le 1.5S$	***Not Applicable***
$S_m^s + S_{bo}^{sX} \le 1.5S$	$S_m^l + S_{bo}^{lX} \le 1.5S$	

Table 4.12.4 – Stress Calculations And Acceptance Criteria For Type 3 Noncircular Vessels (Chamfered Rectangular Cross Section)

Nomenclature For Stress Results	
S_m^s	membrane stress in the short side.
S_{bi}^{sC}, S_{bo}^{sC}	bending stress in the short side at point C on the inside and outside surfaces, respectively.
S_{bi}^{sD}, S_{bo}^{sD}	bending stress in the short side at point D on the inside and outside surfaces, respectively.
S_{bi}^{sX}, S_{bo}^{sX}	bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.
S_m^l	membrane stress in the long side.
S_{bi}^{lA}, S_{bo}^{lA}	bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lB}, S_{bo}^{lB}	bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY}	bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^c	membrane stress in the circular arc between B and C.
S_{bi}^{cBC}, S_{bo}^{cBC}	bending stress in the circular arc between B and C on the inside and outside surfaces, respectively.

Table 4.12.5 – Stress Calculations And Acceptance Criteria For Type 4 Noncircular Vessels (Reinforced Rectangular Cross Section)

Membrane And Bending Stresses – Critical Locations of Maximum Stress
$S_m^s = \dfrac{Php}{2\left(A_1 + t_1 p\right)E_m}$
$S_{bi}^{sC} = -S_{bo}^{sC}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Ppc_i}{24I_{11}E_b}\left[-3H^2 + 2h^2\left(\dfrac{1+\alpha_1^2 k}{1+k}\right)\right]$
$S_{bi}^{sB} = -S_{bo}^{sB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Ph^2 pc_i}{12I_{11}E_b}\left[\dfrac{1+\alpha_1^2 k}{1+k}\right]$
$S_m^l = \dfrac{PHp}{2\left(A_2 + t_2 p\right)E_m}$
$S_{bi}^{lA} = -S_{bo}^{lA}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Ph^2 pc_i}{24I_{21}E_b}\left[-3 + 2\left(\dfrac{1+\alpha_1^2 k}{1+k}\right)\right]$
$S_{bi}^{lB} = -S_{bo}^{lB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Ph^2 pc_i}{12I_{21}E_b}\left[\dfrac{1+\alpha_1^2 k}{1+k}\right]$
Membrane And Bending Stresses – Defined Locations for Stress Calculation
$S_{bi}^{sX} = -S_{bo}^{sX}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Ppc_i}{24I_{11}E_b}\left[-3H^2 + 2h^2\left(\dfrac{1+\alpha_1^2 k}{1+k}\right) + 12X^2\right]$
$S_{bi}^{lY} = -S_{bo}^{lY}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Ph^2 pc_i}{24I_{21}E_b}\left[-3 + 2\left(\dfrac{1+\alpha_1^2 k}{1+k}\right) + \dfrac{12Y^2}{h^2}\right]$
Equation Constants
$k = \dfrac{I_{21}}{I_{11}}\alpha_1$ $\alpha_1 = \dfrac{H_1}{h_1}$

Table 4.12.5 – Stress Calculations And Acceptance Criteria For Type 4 Noncircular Vessels (Reinforced Rectangular Cross Section)

Acceptance Criteria – Critical Locations of Maximum Stress	
$S_m^s \le S$	$S_m^l \le S$
$S_m^s + S_{bi}^{sC} \le 1.5S$	$S_m^l + S_{bi}^{lA} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$	$S_m^l + S_{bo}^{lA} \le 1.5S$
$S_m^s + S_{bi}^{sB} \le 1.5S$	$S_m^l + S_{bi}^{lB} \le 1.5S$
$S_m^s + S_{bo}^{sB} \le 1.5S$	$S_m^l + S_{bo}^{lB} \le 1.5S$
Acceptance Criteria – Defined Locations for Stress Calculation	
$S_m^s + S_{bi}^{sX} \le 1.5S$	$S_m^l + S_{bi}^{lY} \le 1.5S$
$S_m^s + S_{bo}^{sX} \le 1.5S$	$S_m^l + S_{bo}^{lY} \le 1.5S$

Nomenclature For Stress Results	
S_m^s	membrane stress in the short side.
S_{bi}^{sB}, S_{bo}^{sB}	bending stress in the short side at point B on the inside and outside surfaces, respectively.
S_{bi}^{sC}, S_{bo}^{sC}	bending stress in the short side at point C on the inside and outside surfaces, respectively.
S_{bi}^{sX}, S_{bo}^{sX}	bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.
S_m^l	membrane stress in the long side.
S_{bi}^{lB}, S_{bo}^{lB}	bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lA}, S_{bo}^{lA}	bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY}	bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^{st}	membrane stress in the stay bar or plate, as applicable.

Table 4.12.6 – Stress Calculations And Acceptance Criteria For Type 5 Noncircular Vessels (Reinforced Rectangular Cross Section With Chamfered Corners)

Membrane And Bending Stresses – Critical Locations of Maximum Stress
$S_m^s = \dfrac{P\left(R + L_2 + L_{21}\right)}{t_1 E_m}$
$S_{bi}^{sE} = -S_{bo}^{sE}\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1 E_b}\left[M_A + Pp\left\{\dfrac{\left(L_2 + L_{21}\right)^2}{2} + 2R\left(L_2 + L_{21} - L_1 - L_{11}\right)\right\}\right]$
$S_{bi}^{sF} = -S_{bo}^{sF}\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1 E_b}\left[M_A + \dfrac{Pp}{2}\left\{\begin{array}{l} L_2^2 + 2L_2 L_{21} + L_{21}^2 - 2L_1 L_{11} - L_{11}^2 + \\ 2R\left(L_2 + L_{21} - L_1 - L_{11}\right) \end{array}\right\}\right]$
$S_{bi}^{sG} = -S_{bo}^{sG}\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_{11} E_b}\left[M_A + \dfrac{Pp}{2}\left\{\left(L_2 + L_{21}\right)^2 + 2R\left(L_2 + L_{21} - L_1 - L_{11}\right) - \left(L_1 + L_{11}\right)^2\right\}\right]$
$S_m^l = \dfrac{P\left(R + L_1 + L_{11}\right)}{t_2 E_m}$
$S_{bi}^{lA} = -S_{bo}^{lA}\left(\dfrac{c_i}{c_o}\right) = \dfrac{M_A c_i}{I_{21} E_b}$
$S_{bi}^{lB} = -S_{bo}^{lB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_2 E_b}\left[M_A + \dfrac{PpL_2^2}{2}\right]$
$S_{bi}^{lC} = -S_{bo}^{lC}\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_2 E_b}\left[M_A + \dfrac{Pp\left(L_2 + L_{21}\right)^2}{2}\right]$
$S_m^c = \dfrac{P}{t_1 E_m}\left(R + \sqrt{\left(L_2 + L_{21}\right)^2 + \left(L_2 + L_{11}\right)^2}\right)$
$S_{bi}^{cD} = -S_{bo}^{cD}\left(\dfrac{c_i}{c_o}\right) = \dfrac{M_r c_i}{I_1 E_b}$

Table 4.12.6 – Stress Calculations And Acceptance Criteria For Type 5 Noncircular Vessels (Reinforced Rectangular Cross Section With Chamfered Corners)

Membrane And Bending Stresses – Defined Locations for Stress Calculation
$S_{bi}^{sX} = -S_{bo}^{sX}\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_{11}E_b}\left[M_A + \dfrac{Pp}{2}\begin{Bmatrix}\left(L_2+L_{21}\right)^2 + 2R\left(L_2+L_{21}-L_1-L_{11}\right) - \\ \left(L_1+L_{11}\right)^2 + X^2\end{Bmatrix}\right] \qquad X \le L_1$
$S_{bi}^{sX} = -S_{bo}^{sX}\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1E_b}\left[M_A + \dfrac{Pp}{2}\begin{Bmatrix}L_2^2 + 2L_2L_{21} + L_{21}^2 - 2L_1L_{11} - L_{11}^2 + \\ 2R\left(L_2+L_{21}-L_1-L_{11}\right) + X^2\end{Bmatrix}\right] \qquad X > L_1$
$S_{bi}^{lY} = -S_{bo}^{lY}\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_{21}E_b}\left[M_A + \dfrac{PpY^2}{2}\right] \qquad Y \le L_2$
$S_{bi}^{lY} = -S_{bo}^{lY}\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_2E_b}\left[M_A + \dfrac{PpY^2}{2}\right] \qquad Y > L_2$
Equation Constants
$I_1 = \dfrac{pt_1^3}{12}$
$I_2 = \dfrac{pt_2^3}{12}$
$M_A = Pp\dfrac{\begin{bmatrix}-3RL_2\left(4R+\pi L_2\right) - L_{21}\left(12R^2 + 3\pi RL_{21} + 2L_{21}^2\right) + 12RL_{11}^2 - \\ 6L_2L_{21}\left(L_2+L_{21}+\pi R+2L_{11}\right) - 6L_2L_{11}\left(2R+L_2\right) - 6L_{21}L_{11}\left(2R+L_{21}\right) + \\ 6L_1L_{11}\left(2R+L_{11}\right) + 6R^2\left(\pi-2\right)\left(L_1+L_{11}\right) + 4L_{11}^3 - 2L_2^3\left(\dfrac{I_1}{I_{21}}\right) - \\ 2\left(\dfrac{I_1}{I_{11}}\right)\begin{pmatrix}6L_2L_{21}L_1 + 3L_2^2L_1 + 3L_{21}^2L_1 - 6L_1^2L_{11} - 3L_1L_{11}^2 - 2L_1^3 - \\ 6R\left\{L_1^2 - L_2L_1 - L_{21}L_1 + L_1L_{11}\right\}\end{pmatrix}\end{bmatrix}}{\left[6\left\{2L_{21} + 2L_{11} + \pi R + 2L_1\left(\dfrac{I_1}{I_{11}}\right) + 2L_2\left(\dfrac{I_1}{I_{21}}\right)\right\}\right]}$
$M_r = M_A + Pp\left\{\left(L_2+L_{21}\right)\left(\dfrac{L_2+L_{21}}{2} + R\cos\theta\right) + \left(1-\sin\theta\right)\left[R^2 - R\left(L_1+L_{11}+R\right)\right]\right\}$
S_{bi}^{cD} *is a maximum when* $M_r = M_D$ *when* $\theta = \arctan\left(\dfrac{L_1+L_{11}}{L_2+L_{21}}\right)$

Table 4.12.6 – Stress Calculations And Acceptance Criteria For Type 5 Noncircular Vessels (Reinforced Rectangular Cross Section With Chamfered Corners)

Acceptance Criteria – Critical Locations of Maximum Stress		
$S_m^s \le S$	$S_m^l \le S$	$S_m^c \le S$
$S_m^s + S_{bi}^{sE} \le 1.5S$	$S_m^l + S_{bi}^{lA} \le 1.5S$	$S_m^c + S_{bi}^{cD} \le 1.5S$
$S_m^s + S_{bo}^{sE} \le 1.5S$	$S_m^l + S_{bo}^{lA} \le 1.5S$	$S_m^c + S_{bo}^{cD} \le 1.5S$
$S_m^s + S_{bi}^{sF} \le 1.5S$	$S_m^l + S_{bi}^{lB} \le 1.5S$	
$S_m^s + S_{bo}^{sF} \le 1.5S$	$S_m^l + S_{bo}^{lB} \le 1.5S$	
$S_m^s + S_{bi}^{sG} \le 1.5S$	$S_m^l + S_{bi}^{lC} \le 1.5S$	
$S_m^s + S_{bo}^{sG} \le 1.5S$	$S_m^l + S_{bo}^{lC} \le 1.5S$	
Acceptance Criteria – Defined Locations for Stress Calculation		
$S_m^s + S_{bi}^{sX} \le 1.5S$ $S_m^s + S_{bo}^{sX} \le 1.5S$	$S_m^l + S_{bi}^{lY} \le 1.5S$ $S_m^l + S_{bo}^{lY} \le 1.5S$	***Not Applicable***

Nomenclature For Stress Results	
S_m^s	membrane stress in the short side.
S_{bi}^{sE}, S_{bo}^{sE}	bending stress in the short side at point E on the inside and outside surfaces, respectively.
S_{bi}^{sF}, S_{bo}^{sF}	bending stress in the short side at point F on the inside and outside surfaces, respectively.
S_{bi}^{sG}, S_{bo}^{sG}	bending stress in the short side at point G on the inside and outside surfaces, respectively.
S_{bi}^{sX}, S_{bo}^{sX}	bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.
S_m^l	membrane stress in the long side.
S_{bi}^{lA}, S_{bo}^{lA}	bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lB}, S_{bo}^{lB}	bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lC}, S_{bo}^{lC}	bending stress in the long side at point C on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY}	bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^c	membrane stress in the circular arc between B and E.
S_{bi}^{cD}, S_{bo}^{cD}	bending stress in the circular arc at point B on the inside and outside surfaces, respectively.

Table 4.12.7 – Stress Calculations And Acceptance Criteria For Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners)

Membrane And Bending Stresses – Critical Locations of Maximum Stress
$S_m^A = \dfrac{PpL_3}{A_c E_m}$
$S_{bi}^A = -S_{bo}^A \left(\dfrac{c_i}{c_o}\right) = \dfrac{M_A c_i}{I_{21} E_b}$
$S_m^B = \dfrac{PpL_3}{A_c E_m}$
$S_{bi}^B = -S_{bo}^B \left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1 E_b}\left[M_A - V_A \overline{Y}_2 + WL_2^2\right]$
$S_m^C = \dfrac{PpL_3}{A_c E_m}$
$S_{bi}^C = -S_{bo}^C \left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1 E_b}\left[M_A + WK_5^2 - 2L_3 W\overline{Y}_2\right]$
$S_m^M = \dfrac{Pp}{A_c E_m}\left(C_M^2 + \left(L_3 - E_M\right)^2\right)^{0.5} \cos\left(\theta_M - \beta_M\right)$
$S_{bi}^M = -S_{bo}^M \left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1 E_b}\left[M_A + W\left(C_M^2 + C_M V_M + E_M^2 - E_M W_M - L_3\left\{2E_M + t_1 - W_M + 2\overline{Y}_2\right\}\right)\right]$
$S_m^D = \dfrac{PpO_{DE}}{A_c E_m}$
$S_{bi}^D = -S_{bi}^D \left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1 E_b}\left[M_A + W\left(C_3^2 + C_3 V_1 + E_{\theta 1}^2 - E_{\theta 1} W_1 - L_3\left\{2E_{\theta 1} + t_1 - W_1 + 2\overline{Y}_2\right\}\right)\right]$
$S_m^U = \dfrac{PpO_{DE}}{A_c E_m}$
$S_{bi}^U = -S_{bi}^U \left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1 E_b}\left[M_A + W\left(\begin{array}{l}\left\{C_3 + U_{2Y}\right\}^2 + \left\{C_3 + U_{2Y}\right\}V_1 + \left\{E_{\theta 1} + U_{2X}\right\}^2 - \\ \left\{E_{\theta 1} + U_{2X}\right\}W_1 - 2L_3\left\{\overline{Y}_2 + \dfrac{t_1\left(1-\cos\theta_1\right)}{2} + E_{\theta 1} + U_{2X}\right\}\end{array}\right)\right]$

Table 4.12.7 – Stress Calculations And Acceptance Criteria For Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners)

Membrane And Bending Stresses – Critical Locations of Maximum Stress
$S_m^E = \dfrac{PpO_{DE}}{A_c E_m}$
$S_{bi}^E = -S_{bo}^E\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1 E_b}\left[M_A + W\left(\begin{array}{l} C_{E1}^2 + C_{E1}V_1 + C_{E2}^2 - C_{E2}W_1 - \\ 2L_3\left\{\overline{Y}_2 + \dfrac{t_1\left(1-\cos\theta_1\right)}{2} + C_{E2}\right\}\end{array}\right)\right]$
$S_m^N = \dfrac{Pp}{A_c E_m}\left(C_M^2 + O_K^2\right)^{0.5}\cos\left(\theta_N - \beta_N\right)$
$S_{bi}^N = -S_{bo}^N\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1 E_b}\left[M_A + W\left(\begin{array}{l} \left\{L_4 - F_N\right\}^2 + \left\{L_4 - F_N\right\}V_N + \left\{M_1 + G_N\right\}^2 - \\ \left\{M_1 - G_N\right\}W_N - L_3\left\{2\overline{Y}_2 + t_1 + 2M_1 - 2G_N - W_N\right\}\end{array}\right)\right]$
$S_m^F = \dfrac{PpL_4}{A_c E_m}$
$S_{bi}^F = -S_{bo}^F\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1 E_b}\left[M_A + W\left(L_4^2 + L_4 t_1 + M_1^2 - 2L_3 J_2\right)\right]$
$S_m^G = \dfrac{PpL_4}{A_c E_m}$
$S_{bi}^G = -S_{bo}^G\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_1 E_b}\left[M_A + W\left(L_4^2 + L_4 t_1 + \left\{M_1 + L_{11}\right\}^2 - 2L_3\left\{J_2 + L_{11}\right\}\right)\right]$
$S_m^H = \dfrac{PpL_4}{A_c E_m}$
$S_{bi}^H = -S_{bo}^H\left(\dfrac{c_i}{c_o}\right) = \dfrac{c_i}{I_{11} E_b}\left[M_A + W\left(L_4^2 + L_4 t_1 + 2L_4\overline{Y}_1 - L_3^2 - 2L_3\left\{\overline{Y}_2 + \dfrac{t_1}{2}\right\}\right)\right]$

Table 4.12.7 – Stress Calculations And Acceptance Criteria For Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners)

Membrane And Bending Stresses – Defined Locations for Stress Calculation
$S_{bi}^{Y} = -S_{bo}^{Y}\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_{21}E_b}\left[M_A + PpY^2\right] \qquad 0 \le Y \le L_2$
$S_{bi}^{Y} = -S_{bo}^{Y}\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1E_b}\left[M_A + PpY^2\right] \qquad L_2 \le Y \le \left(L_2 + L_{21}\right)$
$S_{bi}^{X} = -S_{bo}^{X}\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_{11}E_b}\left[\begin{array}{l} M_A + \frac{PpX^2}{2} + \\ W\left\{L_4^2 + L_4t_1 + 2L_4\overline{Y}_1 - L_3^2 - 2L_3\left(\overline{Y}_2 + \frac{t_1}{2}\right)\right\}\end{array}\right] \qquad 0 \le X \le L_1$
$S_{bi}^{X} = -S_{bo}^{X}\left(\frac{c_i}{c_o}\right) = \frac{c_i}{I_1E_b}\left[\begin{array}{l} M_A + \frac{PpX^2}{2} + \\ W\left\{L_4^2 + L_4t_1 + 2L_4\overline{Y}_1 - L_3^2 - 2L_3\left(\overline{Y}_2 + \frac{t_1}{2}\right)\right\}\end{array}\right] \qquad L_1 \le X \le \left(L_1 + L_{11}\right)$

Table 4.12.7 – Stress Calculations And Acceptance Criteria For Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners)

Equation Constants
$I_1 = \frac{pt_1^3}{12}$
$M_A = Pp \frac{\left[K_{AB} + K_{BC} + K_{CD} + K_{DE} + K_{EF} + K_{FG} + K_{GH}\right]}{\left[-6\left(\left\{\frac{I_1}{I_{21}}\right\} L_2 + L_{21} + \frac{\pi R}{2} + U_1 + L_{11} + \left\{\frac{I_1}{I_{11}}\right\} L_1\right)\right]}$
$K_{AB} = \left(\frac{I_1}{I_{21}}\right)\left(L_2^3 - D_2 L_2\right)$
$K_{BC} = 3L_2 L_{11} K_5 + L_{21}^3 - D_2 L_{11}$
$K_{CD} = 3R\theta_1\left[K_5^2 + 2R^2 + Rt_1 - L_3\left(S_1 + 2\overline{Y}_2\right)\right] + 3\left[K_5 E_{\theta 1} S_1 + H_{\theta 1} S_1 \left(L_3 - R\right)\right]$
$K_{DE} = 3U_1\left[C_3^2 + C_3 V_1 + E_{\theta 1}^2 - E_{\theta 1} W_1\right] - 6L_3 U_1\left[\overline{Y}_2 + \frac{t_1}{2}\left(1 - \cos\theta_1\right) + E_{\theta 1}\right] +$ $3U_1^2\left[C_3 \cos\theta_1 + \sin\theta_1\left(E_{\theta 1} - L_3\right)\right] + U_1^3$
$K_{EF} = 3R\alpha_{ab}\left[D_3^2 + M_1^2 - 2L_3 J_2 + R^2 + Rt_1\right] + 3G_1 D_3 S_1 + 3F_1 S_1\left(L_3 - M_1\right)$
$K_{FG} = 3L_{11}\left[L_4^2 + L_4 t_1 + M_1^2 - 2L_3 J_2\right] + 3\left[M_1 - L_3\right] L_{11}^2 + L_{11}^3$
$K_{GH} = \left(\frac{I_1}{I_{11}}\right)\left[3L_1\left(L_4^2 + 2L_4\overline{Y} + L_4 t_1 + \left\{M_1 + L_{11}\right\}^2 - 2L_3\left\{J_2 + L_{11}\right\}\right) - 2L_1^3\right]$

Table 4.12.7 – Stress Calculations And Acceptance Criteria For Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners)

Equation Constants	
$A_C = t_1 p$	$S_1 = 2R + t_1$
$A_{DE} = \left[L_4 - \left(L_2 + L_{21} + R\tan\left(\frac{\theta_1}{2}\right)\right)\right]\sin\theta_1$	$U_1 = \sqrt{\left(M_1 - R\right)^2 + \left(N_1 - R\right)^2}$
	$U_2 = 0.5U_1$
$C_3 = L_2 + L_{21} + R\sin\theta_1$	$U_{2X} = U_2 \sin\theta_1$
$C_{E1} = C_3 + N_1 - R$	$U_{2Y} = U_2 \cos\theta_1$
$C_{E2} = E_{\theta 1} + M_1 - R$	$V_1 = t_1 \sin\theta_1$
$C_M = L_2 + L_{21} + R\sin\theta_M$	
$C_N = L_4 - R + R\sin\beta_N$	$V_A = PpL_3$
$D_2 = 6L_4\overline{Y}_2$	$V_M = t_1 \sin\theta_M$
$D_3 = L_4 - R$	$V_N = t_1 \sin\beta_N$
$D_4 = L_1 + L_{11} + R\cos\theta_1$	$W = 0.5Pp$
$E_{\theta 1} = R\left(1 - \cos\theta_1\right)$	$W_1 = t_1 \cos\theta_1$
$E_M = R\left(1 - \cos\theta_M\right)$	$W_M = t_1 \cos\theta_M$
$F_1 = R\left(1 - \sin\theta_1\right)$	$W_N = t_1 \cos\theta_N$
$F_N = R\left(1 - \sin\beta_N\right)$	$\alpha_{ab} = \arctan\left[\frac{L_3}{L_4}\right]$
$G_1 = R\cos\theta_1$	
$G_N = R\cos\beta_N$	$\beta_M = \arctan\left[\frac{C_M}{L_3 - E_{\theta 1}}\right]$
$H_{\theta 1} = R\sin\theta_1$	
$J_2 = \overline{Y}_2 + \frac{t_2}{2} + M_1$	$\beta_N = \arctan\left[\frac{L_4 - R}{L_1 + L_{11}}\right]$
$K_5 = L_2 + L_{21}$	$\theta_1 = \arctan\left[\frac{L_4}{L_3}\right]$
$M_1 = L_3 - \left(L_1 + L_{11}\right)$	
$N_1 = L_4 - \left(L_2 + L_{21}\right)$	$\theta_M = \arctan\left[\frac{-K_5 S_1}{2R^2 - RS_1 - L_3 t_1}\right]$
$O_{DE} = \sqrt{L_3^2 + L_4^2} - A_{DE}$	
$O_K = L_1 + L_{11} + R\cos\beta_N$	$\theta_N = \arctan\left[\frac{C_N}{O_K}\right]$

Table 4.12.7 – Stress Calculations And Acceptance Criteria For Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners)

Acceptance Criteria – Critical Locations of Maximum Stress		
$S_m^A \le S$	$S_m^M \le S$	$S_m^F \le S$
$S_m^A + S_{bi}^A \le 1.5S$	$S_m^M + S_{bi}^M \le 1.5S$	$S_m^F + S_{bi}^F \le 1.5S$
$S_m^A + S_{bo}^A \le 1.5S$	$S_m^M + S_{bo}^M \le 1.5S$	$S_m^F + S_{bo}^F \le 1.5S$
$S_m^B \le S$	$S_m^D \le S$	$S_m^G \le S$
$S_m^B + S_{bi}^B \le 1.5S$	$S_m^D + S_{bi}^D \le 1.5S$	$S_m^G + S_{bi}^G \le 1.5S$
$S_m^B + S_{bo}^B \le 1.5S$	$S_m^D + S_{bo}^D \le 1.5S$	$S_m^G + S_{bo}^G \le 1.5S$
$S_m^C \le S$	$S_m^U \le S$	$S_m^H \le S$
$S_m^C + S_{bi}^C \le 1.5S$	$S_m^U + S_{bi}^U \le 1.5S$	$S_m^H + S_{bi}^H \le 1.5S$
$S_m^C + S_{bo}^C \le 1.5S$	$S_m^U + S_{bo}^U \le 1.5S$	$S_m^H + S_{bo}^H \le 1.5S$
	$S_m^E \le S$	
	$S_m^E + S_{bi}^E \le 1.5S$	
	$S_m^E + S_{bo}^E \le 1.5S$	
	$S_m^N \le S$	
	$S_m^N + S_{bi}^N \le 1.5S$	
	$S_m^N + S_{bo}^N \le 1.5S$	
Acceptance Criteria – Defined Locations for Stress Calculation		
$S_m^Y + S_{bi}^Y \le 1.5S$ $S_m^Y + S_{bo}^Y \le 1.5S$	***Not Applicable***	$S_m^X + S_{bi}^X \le 1.5S$ $S_m^X + S_{bo}^X \le 1.5S$

Table 4.12.7 – Stress Calculations And Acceptance Criteria For Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners)

Nomenclature For Stress Results	
S_m^A	membrane stress at point A.
S_{bi}^A, S_{bo}^A	bending stress at point A on the inside and outside surfaces, respectively.
S_m^B	membrane stress at point B.
S_{bi}^B, S_{bo}^B	bending stress at point B on the inside and outside surfaces, respectively.
S_m^C	membrane stress at point C.
S_{bi}^C, S_{bo}^C	bending stress at point C on the inside and outside surfaces, respectively.
S_m^M	membrane stress at point M.
S_{bi}^M, S_{bo}^M	bending stress at point M on the inside and outside surfaces, respectively.
S_m^D	membrane stress at point D.
S_{bi}^D, S_{bo}^D	bending stress at point D on the inside and outside surfaces, respectively.
S_m^U	membrane stress at point U.
S_{bi}^U, S_{bo}^U	bending stress at point U on the inside and outside surfaces, respectively.
S_m^E	membrane stress at point E.
S_{bi}^E, S_{bo}^E	bending stress at point E on the inside and outside surfaces, respectively.
S_m^N	membrane stress at point N.
S_{bi}^N, S_{bo}^N	bending stress at point N on the inside and outside surfaces, respectively.
S_m^F	membrane stress at point F.
S_{bi}^F, S_{bo}^F	bending stress at point F on the inside and outside surfaces, respectively.
S_m^H	membrane stress at point G.
S_{bi}^G, S_{bo}^G	bending stress at point G on the inside and outside surfaces, respectively.
S_m^H	membrane stress at point H.
S_{bi}^H, S_{bo}^H	bending stress at point H on the inside and outside surfaces, respectively.
S_{bi}^X, S_{bo}^X	bending stress at a point defined by X on the inside and outside surfaces, respectively.
S_{bi}^Y, S_{bo}^Y	bending stress at a point defined by Y on the inside and outside surfaces, respectively.

Table 4.12.8 – Stress Calculations And Acceptance Criteria For Type 7 Noncircular Vessels (Rectangular Cross Section With Single Stay Plate Or Multiple Bars)

Membrane And Bending Stresses – Critical Locations of Maximum Stress	
$S_m^s = \dfrac{Ph}{4t_1E_m}\left[4 - \left(\dfrac{2+K\left(5-\alpha^2\right)}{1+2K}\right)\right]$	
$S_{bi}^{sC} = -S_{bo}^{sC}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbc_i}{24I_1E_b}\left[-3H^2 + 2h^2\left(\dfrac{1+2\alpha^2K}{1+2K}\right)\right]$	
$S_{bi}^{sB} = -S_{bo}^{sB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbh^2c_i}{12I_1E_b}\left[\dfrac{1+2\alpha^2K}{1+2K}\right]$	
$S_m^l = \dfrac{PH}{2t_2E_m}$	
$S_{bi}^{lA} = -S_{bo}^{lA}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbh^2c_i}{12I_2E_b}\left[\dfrac{1+K\left(3-\alpha^2\right)}{1+2K}\right]$	
$S_{bi}^{lB} = -S_{bo}^{lB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbh^2c_i}{12I_2E_b}\left[\dfrac{1+2\alpha^2K}{1+2K}\right]$	
$S_m^{st} = \dfrac{Ph}{2t_3E_{st}}\left[\dfrac{2+K\left(5-\alpha^2\right)}{1+2K}\right]$	*For A Stay Plate*
$S_m^{st} = \dfrac{2Php}{\pi t_3^2E_{st}}\left[\dfrac{2+K\left(5-\alpha^2\right)}{1+2K}\right]$	*For Stay Bars*
Equation Constants	
$I_1 = \dfrac{bt_1^3}{12}$	$K = \dfrac{I_1}{I_2}\alpha$
$I_2 = \dfrac{bt_2^3}{12}$	$\alpha = \dfrac{H}{h}$

Table 4.12.8 – Stress Calculations And Acceptance Criteria For Type 7 Noncircular Vessels (Rectangular Cross Section With Single Stay Plate Or Multiple Bars)

Acceptance Criteria – Critical Locations of Maximum Stress		
$S_m^s \le S$	$S_m^{st} \le S$	$S_m^l \le S$
$S_m^s + S_{bi}^{sC} \le 1.5S$		$S_m^s + S_{bi}^{sA} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$		$S_m^s + S_{bo}^{sA} \le 1.5S$
$S_m^s + S_{bi}^{sB} \le 1.5S$		$S_m^s + S_{bi}^{sB} \le 1.5S$
$S_m^s + S_{bo}^{sB} \le 1.5S$		$S_m^s + S_{bo}^{sB} \le 1.5S$

Nomenclature For Stress Results	
S_m^s	membrane stress in the short side.
S_{bi}^{sB}, S_{bo}^{sB}	bending stress in the short side at point B on the inside and outside surfaces, respectively.
S_{bi}^{sC}, S_{bo}^{sC}	bending stress in the short side at point C on the inside and outside surfaces, respectively.
S_{bi}^{sX}, S_{bo}^{sX}	bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.
S_m^l	membrane stress in the long side.
S_{bi}^{lB}, S_{bo}^{lB}	bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lA}, S_{bo}^{lA}	bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY}	bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^{st}	membrane stress in the stay bar or plate, as applicable.

Table 4.12.9 – Stress Calculations And Acceptance Criteria For Type 8 Noncircular Vessels (Rectangular Cross Section With Double Stay Plate Or Multiple Bars)

<table>
<tr><th colspan="2">Membrane And Bending Stresses – Critical Locations of Maximum Stress</th></tr>
<tr><td colspan="2">

$$S_m^s = \frac{Ph}{2t_1E_m}\left[3-\left(\frac{6+K\left(11-\alpha^2\right)}{3+5K}\right)\right]$$

$$S_{bi}^{sC} = -S_{bo}^{sC}\left(\frac{c_i}{c_o}\right) = \frac{Pbc_i}{24I_1E_b}\left[-3H^2+2h^2\left(\frac{3+5\alpha^2K}{3+5K}\right)\right]$$

$$S_{bi}^{sB} = -S_{bo}^{sB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2c_i}{12I_1E_b}\left[\frac{3+5\alpha^2K}{3+5K}\right]$$

$$S_m^l = \frac{PH}{2t_2E_m}$$

$$S_{bi}^{lA} = -S_{bo}^{lA}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2c_i}{12I_2E_b}\left[\frac{3+K\left(6-\alpha^2\right)}{3+5K}\right]$$

$$S_{bi}^{lB} = -S_{bo}^{lB}\left(\frac{c_i}{c_o}\right) = \frac{Pbh^2c_i}{12I_2E_b}\left[\frac{3+5\alpha^2K}{3+5K}\right]$$

$$S_m^{st} = \frac{Ph}{2t_4E_{st}}\left[\frac{6+K\left(11-\alpha^2\right)}{3+5K}\right] \qquad \textit{For A Stay Plate}$$

$$S_m^{st} = \frac{2Php}{\pi t_4^2E_{st}}\left[\frac{6+K\left(11-\alpha^2\right)}{3+5K}\right] \qquad \textit{For Stay Bars}$$

</td></tr>
<tr><th colspan="2">Equation Constants</th></tr>
<tr><td>

$$I_1 = \frac{bt_1^3}{12}$$

$$I_2 = \frac{bt_2^3}{12}$$

</td><td>

$$K = \frac{I_1}{I_2}\alpha$$

$$\alpha = \frac{H}{h}$$

</td></tr>
</table>

Table 4.12.9 – Stress Calculations And Acceptance Criteria For Type 8 Noncircular Vessels (Rectangular Cross Section With Double Stay Plate Or Multiple Bars)

Acceptance Criteria - Critical Locations of Maximum Stress		
$S_m^s \le S$	$S_m^{st} \le S$	$S_m^l \le S$
$S_m^s + S_{bi}^{sC} \le 1.5S$		$S_m^l + S_{bi}^{lA} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$		$S_m^l + S_{bo}^{lA} \le 1.5S$
$S_m^s + S_{bi}^{sB} \le 1.5S$		$S_m^l + S_{bi}^{lB} \le 1.5S$
$S_m^s + S_{bo}^{sB} \le 1.5S$		$S_m^l + S_{bo}^{lB} \le 1.5S$

Nomenclature For Stress Results	
S_m^s	membrane stress in the short side.
S_{bi}^{sB}, S_{bo}^{sB}	bending stress in the short side at point B on the inside and outside surfaces, respectively.
S_{bi}^{sC}, S_{bo}^{sC}	bending stress in the short side at point C on the inside and outside surfaces, respectively.
S_{bi}^{sX}, S_{bo}^{sX}	bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.
S_m^l	membrane stress in the long side.
S_{bi}^{lB}, S_{bo}^{lB}	bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lA}, S_{bo}^{lA}	bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY}	bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^{st}	membrane stress in the stay bar or plate, as applicable.

Table 4.12.10 – Stress Calculations And Acceptance Criteria For Type 9 Noncircular Vessels (Obround Cross Section)

Membrane And Bending Stresses – Critical Locations of Maximum Stress
$S_m^{cB} = \dfrac{PR}{t_1 E_m}$ $S_{bi}^{cB} = -S_{bo}^{cB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{PbL_2c_i}{6I_1E_b}\left[3L_2 - \dfrac{C_1}{A}\right]$ $S_m^{cC} = \dfrac{P\left(R+L_2\right)}{t_1E_m}$ $S_{bi}^{cC} = -S_{bo}^{cC}\left(\dfrac{c_i}{c_o}\right) = \dfrac{PbL_2c_i}{6I_1E_b}\left[3\left(L_2+2R\right) - \dfrac{C_1}{A}\right]$ $S_m^{l} = \dfrac{PR}{t_1E_m}$ $S_{bi}^{lA} = -S_{bo}^{lA}\left(\dfrac{c_i}{c_o}\right) = \dfrac{PbL_2C_1c_i}{6AI_2E_b}\left(\dfrac{t_2}{2}\right)$ $S_{bi}^{lB} = -S_{bo}^{lB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{PbL_2c_i}{6I_2E_b}\left[3L_2 - \dfrac{C_1}{A}\right]$
Membrane And Bending Stresses – Defined Locations for Stress Calculation
$S_{bi}^{lY} = -S_{bo}^{lY}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pc_i}{I_2E_b}\left[\dfrac{-L_2C_1}{6A} + \dfrac{Y^2}{2}\right]$
Equation Constants
$I_1 = \dfrac{bt_1^3}{12}$ $I_2 = \dfrac{bt_2^3}{12}$ $A = R\left[2\left(\dfrac{L_2}{R}\right) + \pi\left(\dfrac{I_2}{I_1}\right)\right]$ $C_1 = R^2\left[2\left(\dfrac{L_2}{R}\right)^2 + 3\pi\left(\dfrac{L_2}{R}\right)\left(\dfrac{I_2}{I_1}\right) + 12\left(\dfrac{I_2}{I_1}\right)\right]$

Table 4.12.10 – Stress Calculations And Acceptance Criteria For Type 9 Noncircular Vessels (Obround Cross Section)

Acceptance Criteria – Critical Locations of Maximum Stress	
$S_m^{cB} \le S$	$S_m^{l} \le S$
$S_m^{cB} + S_{bi}^{cB} \le 1.5S$	$S_m^{l} + S_{bi}^{lA} \le 1.5S$
$S_m^{cB} + S_{bo}^{cB} \le 1.5S$	$S_m^{l} + S_{bo}^{lA} \le 1.5S$
$S_m^{cC} \le S$	$S_m^{l} + S_{bi}^{lB} \le 1.5S$
$S_m^{cC} + S_{bi}^{cC} \le 1.5S$	$S_m^{l} + S_{bo}^{lB} \le 1.5S$
$S_m^{cC} + S_{bo}^{cC} \le 1.5S$	
Acceptance Criteria – Defined Locations for Stress Calculation	
Not Applicable	$S_m^{l} + S_{bi}^{lY} \le 1.5S$ $S_m^{l} + S_{bo}^{lY} \le 1.5S$
Nomenclature For Stress Results	
S_m^{cB}	membrane stress in the circular arc at point B.
S_{bi}^{cB}, S_{bo}^{cB}	bending stress in the circular arc at point B on the inside and outside surfaces, respectively.
S_m^{cC}	membrane stress in the circular arc at point C.
S_{bi}^{cC}, S_{bo}^{cC}	bending stress in the circular arc at point C on the inside and outside surfaces, respectively.
S_m^{l}	membrane stress in the long side.
S_{bi}^{lB}, S_{bo}^{lB}	bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lA}, S_{bo}^{lA}	bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY}	bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^{st}	membrane stress in stay bar or plate, as applicable.

Table 4.12.11 – Stress Calculations And Acceptance Criteria For Type 10 Noncircular Vessels (Reinforced Obround Cross Section)

Membrane And Bending Stresses – Critical Locations of Maximum Stress
$S_m^{cB} = \dfrac{PRp}{\left(A_1 + pt_1\right)E_m}$
$S_{bi}^{cB} = -S_{bo}^{cB}\left(\dfrac{c_i^c}{c_o^c}\right) = \dfrac{PpL_2c_i^c}{6I_{11}E_b}\left[3L_2 - \dfrac{C_2}{A_3}\right]$
$S_m^{cC} = \dfrac{P\left(R + L_2\right)p}{\left(A_1 + pt_1\right)E_m}$
$S_{bi}^{cC} = -S_{bo}^{cC}\left(\dfrac{c_i^c}{c_o^c}\right) = \dfrac{PpL_2c_i^c}{6I_{11}E_b}\left[3\left(L_2 + 2r\right) - \dfrac{C_2}{A_3}\right]$
$S_m^{l} = \dfrac{PRp}{\left(A_1 + pt_1\right)E_m}$
$S_{bi}^{lA} = -S_{bo}^{lA}\left(\dfrac{c_i^l}{c_o^l}\right) = \dfrac{PL_2pc_i^l}{6I_{11}E_b}\left[\dfrac{-C_2}{A_3}\right]$
$S_{bi}^{lB} = -S_{bo}^{lB}\left(\dfrac{c_i^l}{c_o^l}\right) = \dfrac{PL_2pc_i^l}{6I_{11}E_b}\left[3L_2 - \dfrac{C_2}{A_3}\right]$

Table 4.12.11 – Stress Calculations And Acceptance Criteria For Type 10 Noncircular Vessels (Reinforced Obround Cross Section)

Membrane And Bending Stresses – Defined Locations for Stress Calculation	
$S_{bi}^{lY} = -S_{bo}^{lY}\left(\frac{c_i^l}{c_o^l}\right) = \frac{Ppc_i^l}{I_{11}E_b}\left[\frac{-L_2C_2}{6A_3} + \frac{Y^2}{2}\right]$	
Equation Constants	
$A_3 = r\left[2\left(\frac{L_2}{r}\right) + \pi\right]$	$C_2 = r^2\left[2\left(\frac{L_2}{r}\right)^2 + 3\pi\left(\frac{L_2}{r}\right) + 12\right]$
Acceptance Criteria – Critical Locations of Maximum Stress	
$S_m^{cB} \le S$	$S_m^l \le S$
$S_m^{cB} + S_{bi}^{cB} \le 1.5S$	$S_m^l + S_{bi}^{lA} \le 1.5S$
$S_m^{cB} + S_{bo}^{cB} \le 1.5S$	$S_m^l + S_{bo}^{lA} \le 1.5S$
$S_m^{cC} \le S$	$S_m^l + S_{bi}^{lB} \le 1.5S$
$S_m^{cC} + S_{bi}^{cC} \le 1.5S$	$S_m^l + S_{bo}^{lB} \le 1.5S$
$S_m^{cC} + S_{bo}^{cC} \le 1.5S$	
Acceptance Criteria – Defined Locations for Stress Calculation	
Not Applicable	$S_m^l + S_{bi}^{lY} \le 1.5S$ $S_m^l + S_{bo}^{lY} \le 1.5S$
Nomenclature For Stress Results	

Table 4.12.11 – Stress Calculations And Acceptance Criteria For Type 10 Noncircular Vessels (Reinforced Obround Cross Section)

S_m^{cB}	membrane stress in the circular arc at point B.
S_{bi}^{cB}, S_{bo}^{cB}	bending stress in the circular arc at point B on the inside and outside surfaces, respectively.
S_m^{cC}	membrane stress in the circular arc at point C.
S_{bi}^{cC}, S_{bo}^{cC}	bending stress in the circular arc at point C on the inside and outside surfaces, respectively.
S_m^{l}	membrane stress in the long side.
S_{bi}^{lB}, S_{bo}^{lB}	bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lA}, S_{bo}^{lA}	bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY}	bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^{st}	membrane stress in stay bar or plate, as applicable.

Table 4.12.12 – Stress Calculations And Acceptance Criteria For Type 11 Noncircular Vessels (Obround Cross Section With Single Stay Plate Or Multiple Bars)

Membrane And Bending Stresses - Defined Locations for Stress Calculation
$S_m^{cB} = \dfrac{PR}{t_1 E_m}$
$S_{bi}^{cB} = -S_{bo}^{cB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{PbL_2 c_i}{2I_1 A E_b}\left[F\left(B - AL_2\right) - \dfrac{C_1}{3} + AL_2\right]$
$S_m^{cC} = \dfrac{P}{2t_1 E_m}\left[2\left(R + L_2\right) - L_2 F\right]$
$S_{bi}^{cC} = -S_{bo}^{cC}\left(\dfrac{c_i}{c_o}\right) = \dfrac{PbL_2 c_i}{2I_1 A E_b}\left[F\left(B - AL_2 - AR\right) - \dfrac{C_1}{3} + A\left(L_2 + 2R\right)\right]$
$S_m^{l} = \dfrac{PR}{t_2 E_m}$
$S_{bi}^{lA} = -S_{bo}^{lA}\left(\dfrac{c_i}{c_o}\right) = \dfrac{PbL_2 c_i}{2I_2 A E_b}\left[BF - \dfrac{C_1}{3}\right]$
$S_{bi}^{lB} = -S_{bo}^{lB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{PbL_2 c_i}{2I_2 A E_b}\left[F\left(B - AL_2\right) - \dfrac{C_1}{3} + AL_2\right]$
$S_m^{st} = \dfrac{PL_2 F}{t_3 E_{st}}$ *For A Stay Plate*
$S_m^{st} = \dfrac{4PL_2 Fp}{\pi t_3^2 E_{st}}$ *For Stay Bars*

Table 4.12.12 – Stress Calculations And Acceptance Criteria For Type 11 Noncircular Vessels (Obround Cross Section With Single Stay Plate Or Multiple Bars)

Equation Constants	
$I_1 = \dfrac{bt_1^3}{12}$	$F = \dfrac{\left(3AD_1 - 2BC_1\right)}{\left(AE_1 - 6B^2\right)}$
$I_2 = \dfrac{bt_2^3}{12}$	$D_1 = R^3 \left[\left(\dfrac{L_2}{R}\right)^3 + 2\pi\left(\dfrac{L_2}{R}\right)^2\left(\dfrac{I_2}{I_1}\right) + 12\left(\dfrac{L_2}{R}\right)\left(\dfrac{I_2}{I_1}\right) + 2\pi\left(\dfrac{I_2}{I_1}\right) \right]$
$A = R\left[2\left(\dfrac{L_2}{R}\right) + \pi\left(\dfrac{I_2}{I_1}\right)\right]$	
$B = R^2\left[\left(\dfrac{L_2}{R}\right)^2 + \pi\left(\dfrac{L_2}{R}\right)\left(\dfrac{I_2}{I_1}\right) + 2\left(\dfrac{I_2}{I_1}\right)\right]$	$E_1 = R^3 \left[4\left(\dfrac{L_2}{R}\right)^3 + 6\pi\left(\dfrac{L_2}{R}\right)^2\left(\dfrac{I_2}{I_1}\right) + 24\left(\dfrac{L_2}{R}\right)\left(\dfrac{I_2}{I_1}\right) + 3\pi\left(\dfrac{I_2}{I_1}\right) \right]$
$C_1 = R^2\left[2\left(\dfrac{L_2}{R}\right)^2 + 3\pi\left(\dfrac{L_2}{R}\right)\left(\dfrac{I_2}{I_1}\right) + 12\left(\dfrac{I_2}{I_1}\right)\right]$	

Acceptance Criteria - Critical Locations of Maximum Stress		
$S_m^{cB} \le S$	$S_m^{st} \le S$	$S_m^{l} \le S$
$S_m^{cB} + S_{bi}^{cB} \le 1.5S$		$S_m^{l} + S_{bi}^{lA} \le 1.5S$
$S_m^{cB} + S_{bo}^{cB} \le 1.5S$		$S_m^{l} + S_{bo}^{lA} \le 1.5S$
$S_m^{cC} \le S$		$S_m^{l} + S_{bi}^{lB} \le 1.5S$
$S_m^{cC} + S_{bi}^{cC} \le 1.5S$		$S_m^{l} + S_{bo}^{lB} \le 1.5S$
$S_m^{cC} + S_{bo}^{cC} \le 1.5S$		

Table 4.12.12 – Stress Calculations And Acceptance Criteria For Type 11 Noncircular Vessels (Obround Cross Section With Single Stay Plate Or Multiple Bars)

Nomenclature For Stress Results	
S_m^{cB}	membrane stress in the circular arc at point B.
S_{bi}^{cB}, S_{bo}^{cB}	bending stress in the circular arc at point B on the inside and outside surfaces, respectively.
S_m^{cC}	membrane stress in the circular arc at point C.
S_{bi}^{cC}, S_{bo}^{cC}	bending stress in the circular arc at point C on the inside and outside surfaces, respectively.
S_m^{l}	membrane stress in the long side.
S_{bi}^{lB}, S_{bo}^{lB}	bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lA}, S_{bo}^{lA}	bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY}	bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^{st}	membrane stress in stay bar or plate, as applicable.

Table 4.12.13 – Stress Calculations And Acceptance Criteria For Type 12 Noncircular Vessels (Circular Cross Section With Single Stay Plate)

<table>
<tr><th colspan="2">Membrane And Bending Stresses – Critical Locations of Maximum Stress</th></tr>
<tr><th>Equal Pressure</th><th>Unequal Pressure</th></tr>
<tr><td>

$$S_m = \frac{P_1 R}{t_1 E_m}$$

$$S_{bi} = -S_{bo}\left(\frac{c_i}{c_o}\right) = \frac{bc_i}{I_1 E_b}\left[\frac{2P_1 t_1^2}{3\left(\pi^2 - 8\right)}\right]$$

$$S_m^{st} = \frac{2\pi P_1 t_1^2}{3Rt_3\left(\pi^2 - 8\right)E_m}$$

</td><td>

$$S_m = \frac{PR}{t_1 E_m}$$

$$S_{bi} = -S_{bo}\left(\frac{c_i}{c_o}\right) = \frac{bc_i}{3I_1 E_b}\left[P_1\left(\frac{2t_1^2}{\pi^2 - 8}\right) + \frac{3R^2\left(P_1 - P_2\right)}{6 + \left(\frac{t_3}{t_1}\right)^3}\right]$$

$$S_m^{st} = \frac{\pi t_1^2\left(P_1 + P_2\right)}{3Rt_3\left(\pi^2 - 8\right)E_m}$$

$$S_{bi}^{st} = -S_{bo}^{st}\left(\frac{c_i}{c_o}\right) = \frac{J_1\left(P_1 - P_2\right)L_v^2 bc_i}{I_3 E_b} \qquad For\ L_1 \le 2R$$

$$S_{bi}^{st} = -S_{bo}^{st}\left(\frac{c_i}{c_o}\right) = \frac{J_1\left(P_1 - P_2\right)4R^2 bc_i}{I_3 E_b} \qquad For\ L_1 > 2R$$

</td></tr>
<tr><th colspan="2">Equation Constants</th></tr>
<tr><td colspan="2">

$$I_1 = \frac{bt_1^3}{12}$$

$$I_3 = \frac{bt_3^3}{12}$$

$$J_1 = \min\left[\begin{pmatrix} -0.055314 + 0.14237\left(R_h\right) - 0.041311\left(R_h\right)^2 + \\ 0.0050644\left(R_h\right)^3 - 0.00021145\left(R_h\right)^4 \end{pmatrix},\ 5.0\right]$$

$$R_h = \max\left[\frac{L_v}{2R},\ \frac{2R}{L_v}\right]$$

</td></tr>
<tr><th colspan="2">Acceptance Criteria - Critical Locations of Maximum Stress</th></tr>
<tr><th>Equal Pressure</th><th>Unequal Pressure</th></tr>
<tr><td>

$S_m \le S$

$S_m + S_{bi} \le 1.5S$

$S_m + S_{bo} \le 1.5S$

$S_m^{st} \le S$

</td><td>

$S_m \le S$

$S_m + S_{bi} \le 1.5S$

$S_m + S_{bo} \le 1.5S$

$S_m^{st} \le S$

$S_m^{st} + S_{bi}^{st} \le 1.5S$

$S_m^{st} + S_{bo}^{st} \le 1.5S$

</td></tr>
</table>

Table 4.12.13 – Stress Calculations And Acceptance Criteria For Type 12 Noncircular Vessels (Circular Cross Section With Single Stay Plate)

Nomenclature For Stress Results	
S_m	membrane stress in the pipe.
S_{bi}, S_{bo}	bending stress in the pipe.
S_m^{st}	membrane stress in stay plate, as applicable.
S_{bi}^{st}, S_{bo}^{st}	bending stress in the stay plate on the inside and outside surfaces, respectively.

Table 4.12.14 – Effective Width Coefficient

Material	Effective Width Coefficient Δ	
	$\sqrt{psi}$	$\sqrt{kPa}$
Carbon Steel	6000	15754
Austenitic Stainless Steel	5840	15334
Ni-Cr-Fe	6180	16229
Ni-Fe-Cr	6030	15834
Aluminum	3560	9348
Nickel Copper	5720	15021
Unalloyed Titanium	4490	11789

Table 4.12.15 – Compressive Stress Calculations

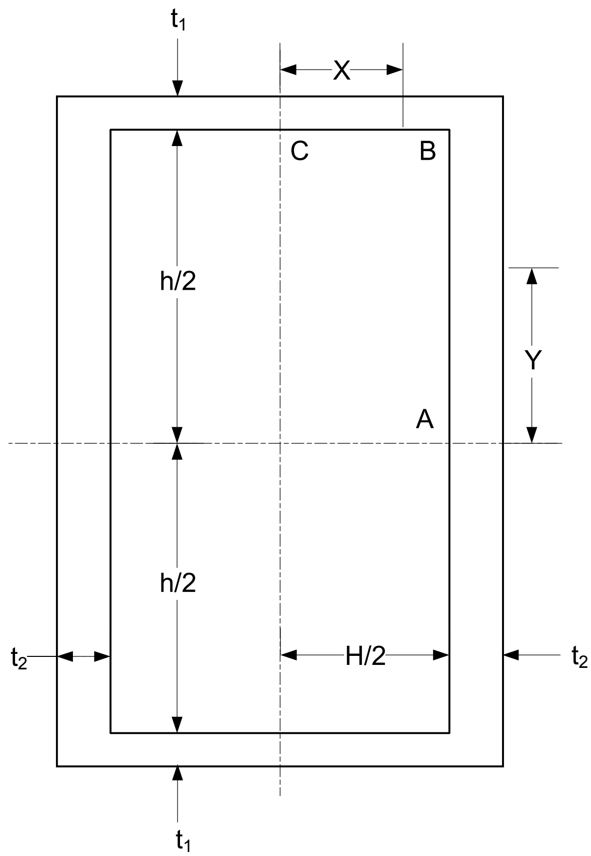
Short Side Plates	Long Side Plates	End Plates
$S_{mA}=\dfrac{P_e hH}{2\left(t_1 H+t_2 h\right)}$	$S_{mA}=\dfrac{P_e hH}{2\left(t_1 H+t_2 h\right)}$	$S_{mA}=\dfrac{P_e HL_v}{2\left(t_2 L_v+t_5 H\right)}$
$S_{mB}=\dfrac{P_e h}{2t_1}$	$S_{mB}=\dfrac{P_e H}{2t_2}$	$S_{mB}=\dfrac{P_e hL_v}{2\left(t_1 L_v+t_5 h\right)}$
$S_{crA}^{*}=\dfrac{\pi^2 E_y}{12\left(1-\nu^2\right)}\left(\dfrac{t_1}{H}\right)^2 K_A$	$S_{crA}^{*}=\dfrac{\pi^2 E_y}{12\left(1-\nu^2\right)}\left(\dfrac{t_2}{h}\right)^2 K_A$	$S_{crA}^{*}=\dfrac{\pi^2 E_y}{12\left(1-\nu^2\right)}\left(\dfrac{t_5}{H}\right)^2 K_A$
$S_{crA}^{**}=S_y-\dfrac{S_y^2}{4S_{crA}^{*}}$	$S_{crA}^{**}=S_y-\dfrac{S_y^2}{4S_{crA}^{*}}$	$S_{crA}^{**}=S_y-\dfrac{S_y^2}{4S_{crA}^{*}}$
$S_{crB}^{*}=\dfrac{\pi^2 E_y}{12\left(1-\nu^2\right)}\left(\dfrac{t_1}{L_v}\right)^2 K_B$	$S_{crB}^{*}=\dfrac{\pi^2 E_y}{12\left(1-\nu^2\right)}\left(\dfrac{t_2}{L_v}\right)^2 K_B$	$S_{crB}^{*}=\dfrac{\pi^2 E_y}{12\left(1-\nu^2\right)}\left(\dfrac{t_5}{h}\right)^2 K_B$
$S_{crB}^{**}=S_y-\dfrac{S_y^2}{4S_{crB}^{*}}$	$S_{crB}^{**}=S_y-\dfrac{S_y^2}{4S_{crB}^{*}}$	$S_{crB}^{**}=S_y-\dfrac{S_y^2}{4S_{crB}^{*}}$
$K_A=K_A\left(x=\dfrac{L_v}{H}\right)$	$K_A=K_A\left(x=\dfrac{L_v}{h}\right)$	$K_A=K_A\left(x=\dfrac{h}{H}\right)$
$K_B=K_B\left(x=\dfrac{H}{L_v}\right)$	$K_B=K_B\left(x=\dfrac{h}{L_v}\right)$	$K_B=K_B\left(x=\dfrac{H}{h}\right)$

Nomenclature For Stress Results	
S_{mA}	compressive stress applied to the short edge of the side panels due to external pressure on the end plates.
S_{mB}	compressive stress applied to the long edge of the side panels due to external pressure on the end plates.
S_{crA}	plate buckling stress when the panel is subjected to stress on the short edge.
S_{crB}	plate buckling stress when the panel is subjected to stress on the long edge.

Table 4.12.15 – Compressive Stress Calculations

Notes
Notes: 1. The equations for K_A and K_B are: $K_A(x) = \max\left[\left(5.2184 + \frac{1.4597}{x} + \frac{0.30384}{x^2}\right),\ 5.50\right]$ (4.12.58) $K_B(x) = 5.2184 + \frac{1.4597}{x} + \frac{0.30384}{x^2} \quad for \quad x \geq 0.258$ (4.12.59) $K_B(x) = 1.0 \quad for \quad x < 0.258$ (4.12.60) 2. The membrane equations for S_{mA} in this table apply to vessels where the long plate thicknesses are equal. If the thicknesses are not equal, replace $2t_2$ with $(t_2 + t_{22})$ in the calculations. 3. The membrane equation S_{mB} in this table for the long side plate applies to vessels where the long plate thicknesses are equal. If the thicknesses are not equal, the membrane stress for the long side plates shall be determined in accordance with Table 4.12.3. 4. Note in the above nomenclature, $K_A = K_A\left(x = \frac{H}{L_v}\right)$ is defined as computing K_A using the function $K_A(x)$ evaluated at $x = \frac{H}{L_v}$.

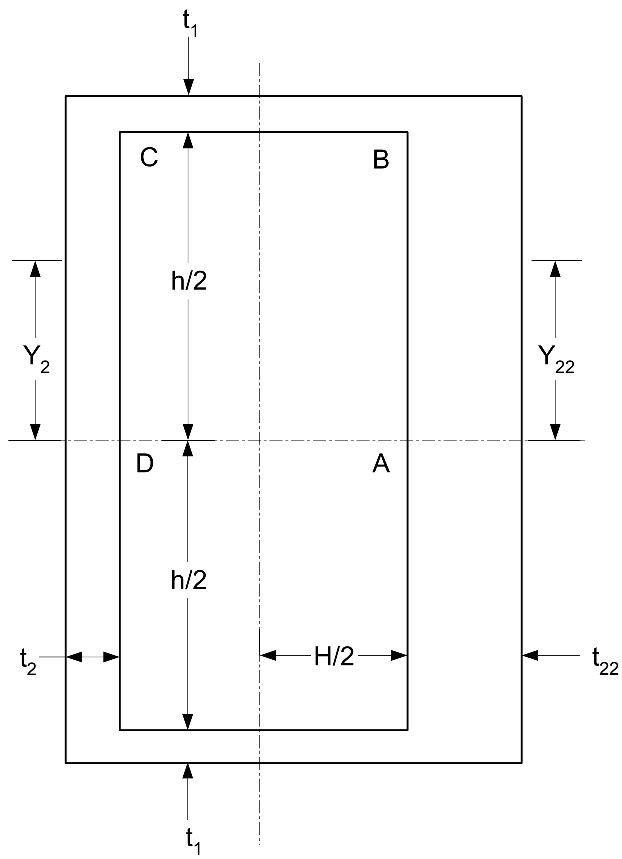
4.12.13 Figures



Notes:

1. Critical Locations of Maximum Stress are defined at points A, B, and C.
2. Defined Locations for Stress Calculations are determined using variables X and Y.

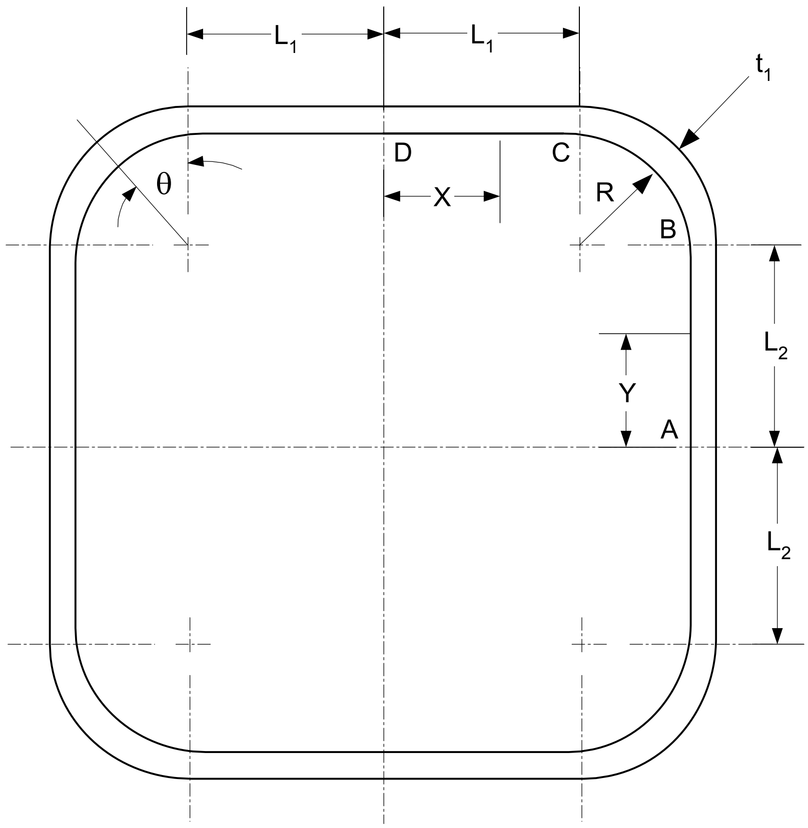
Figure 4.12.1 – Type 1 Noncircular Vessels (Rectangular Cross Section)



Notes:

1. Critical Locations of Maximum Stress are defined at points A, B, C, and D.
2. Defined Locations for Stress Calculations are determined using variables Y_2 and Y_{22}.

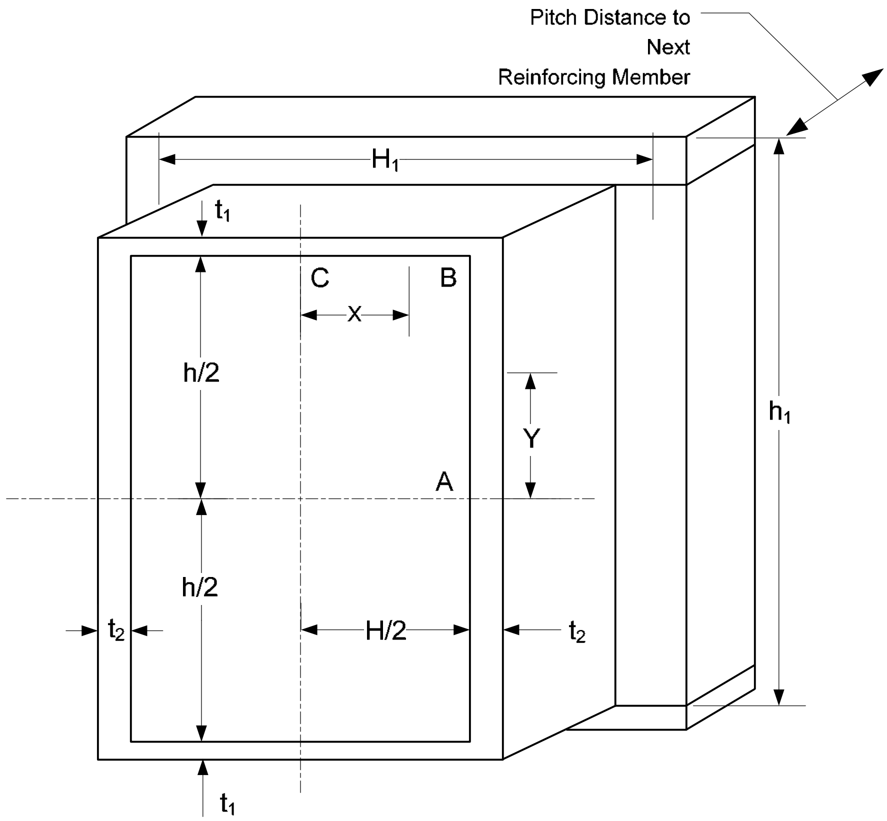
Figure 4.12.2 – Type 2 Noncircular Vessels (Rectangular Cross Section With Unequal Side Plate Thicknesses)



Notes:

1. Critical Locations of Maximum Stress are defined at points A, B, C, and D.
2. Defined Locations for Stress Calculations are determined using variables X and Y.

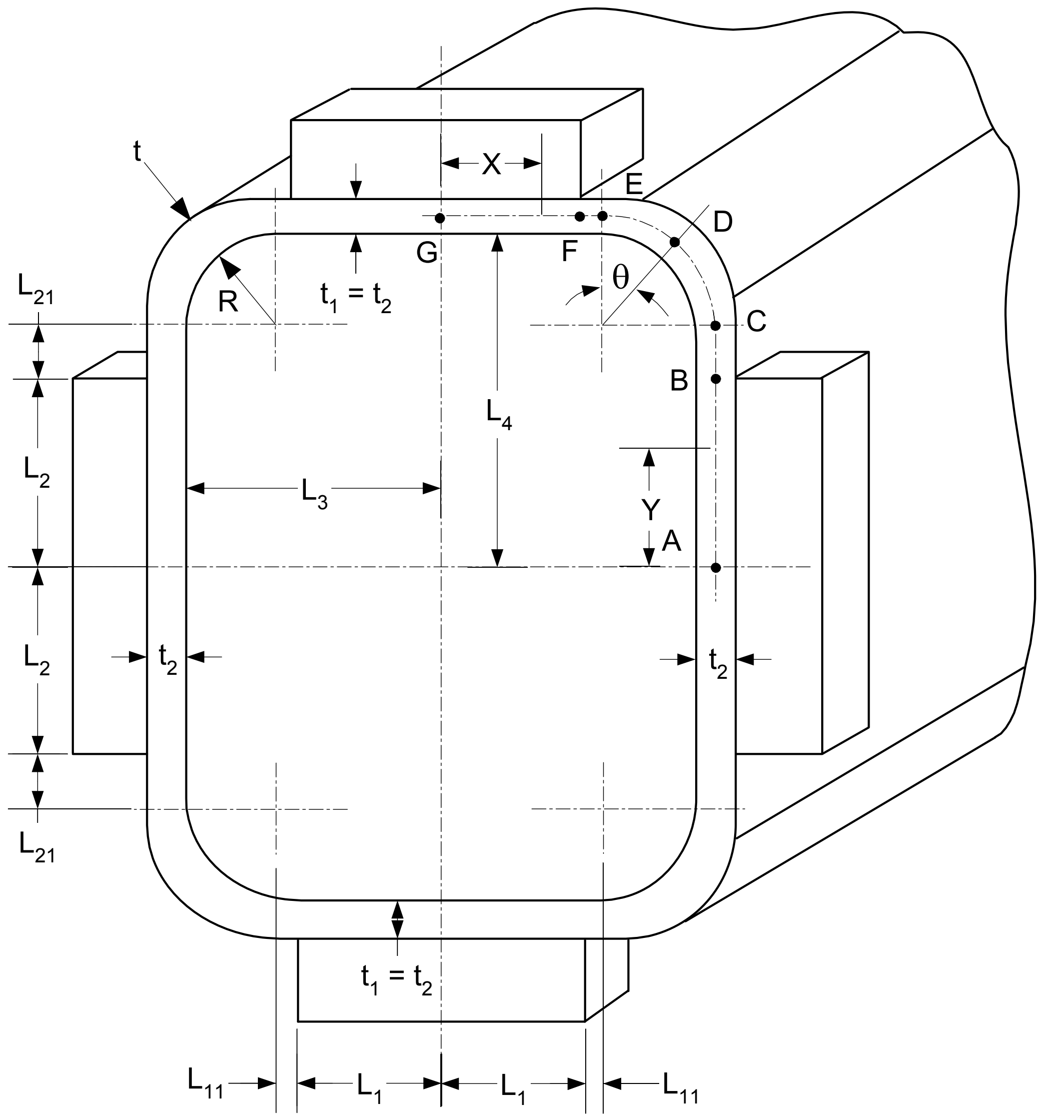
Figure 4.12.3 – Type 3 Noncircular Vessels (Chamfered Rectangular Cross Section)



Notes:

1. Critical Locations of Maximum Stress are defined at points A, B, and C.
2. Defined Locations for Stress Calculations are determined using variables X and Y.

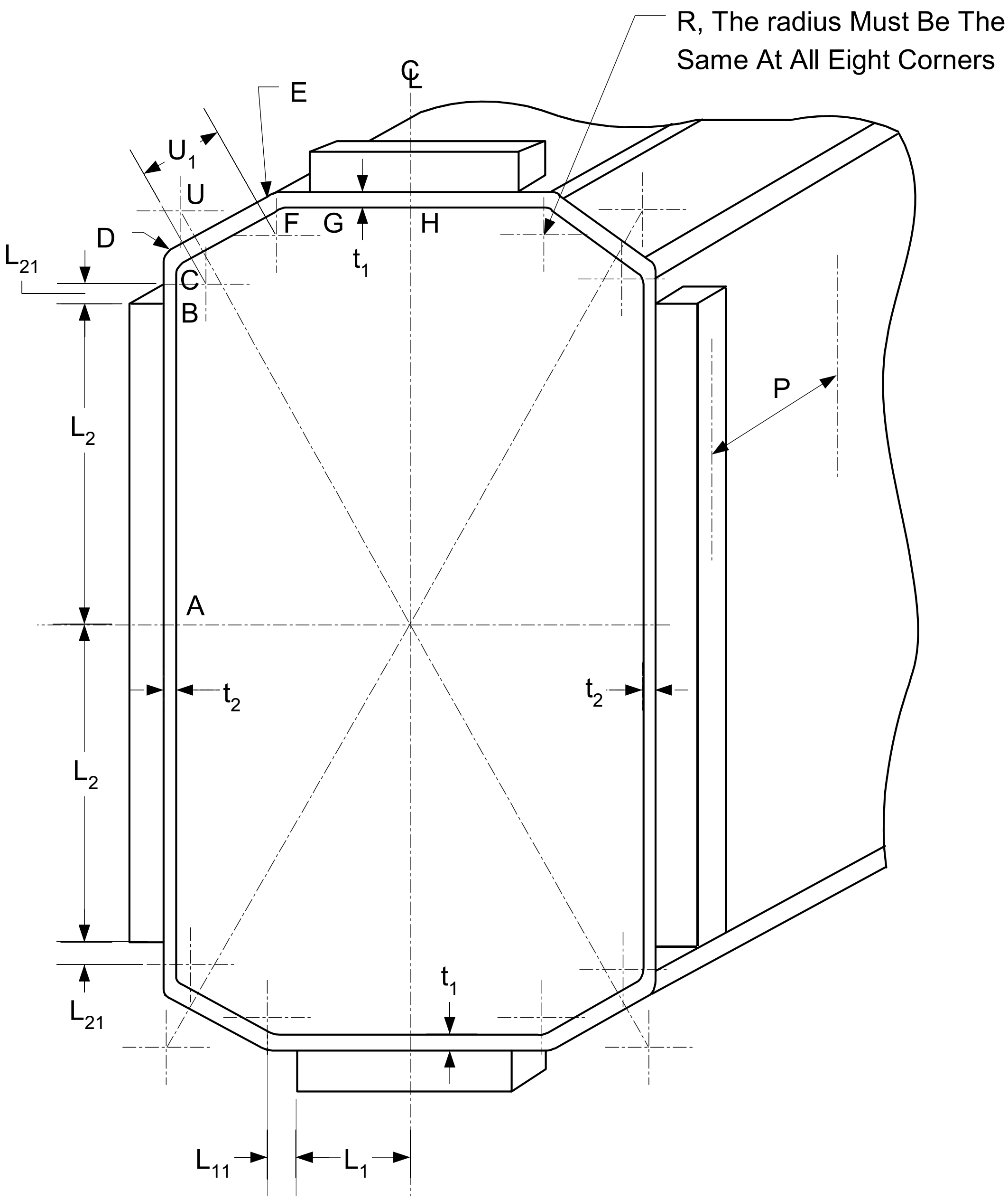
Figure 4.12.4 – Type 4 Noncircular Vessels
(Reinforced Rectangular Cross Section)



Notes:

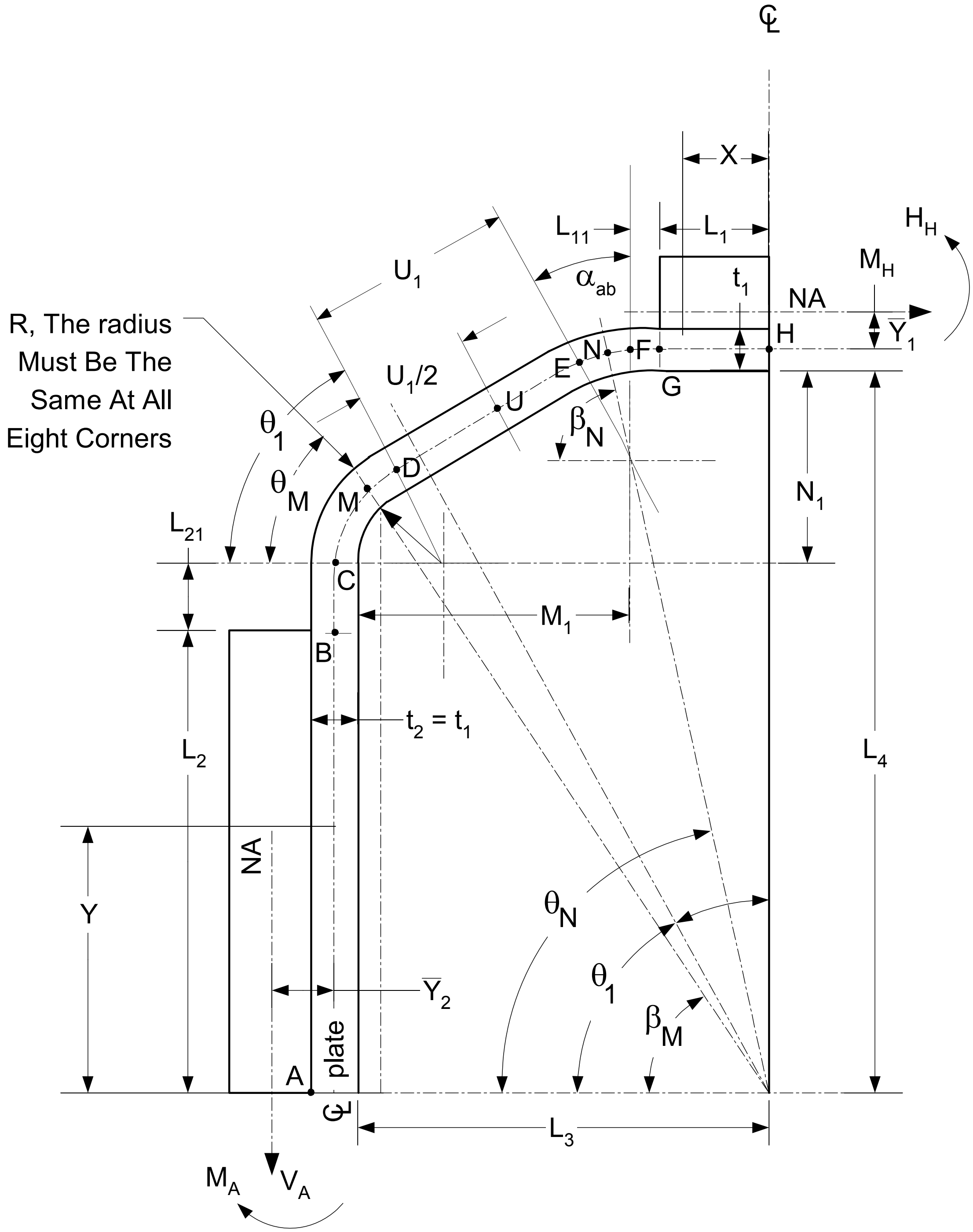
1. Critical Locations of Maximum Stress are defined at points A, B, C, E, F, and G.
2. Defined Locations for Stress Calculations are determined using variables X and Y.

Figure 4.12.5 – Type 5 Noncircular Vessels (Reinforced Rectangular Cross Section With Chamfered Corners)



Note: Critical Locations of Maximum Stress are defined at points A, B, C, E, F, G, and H.

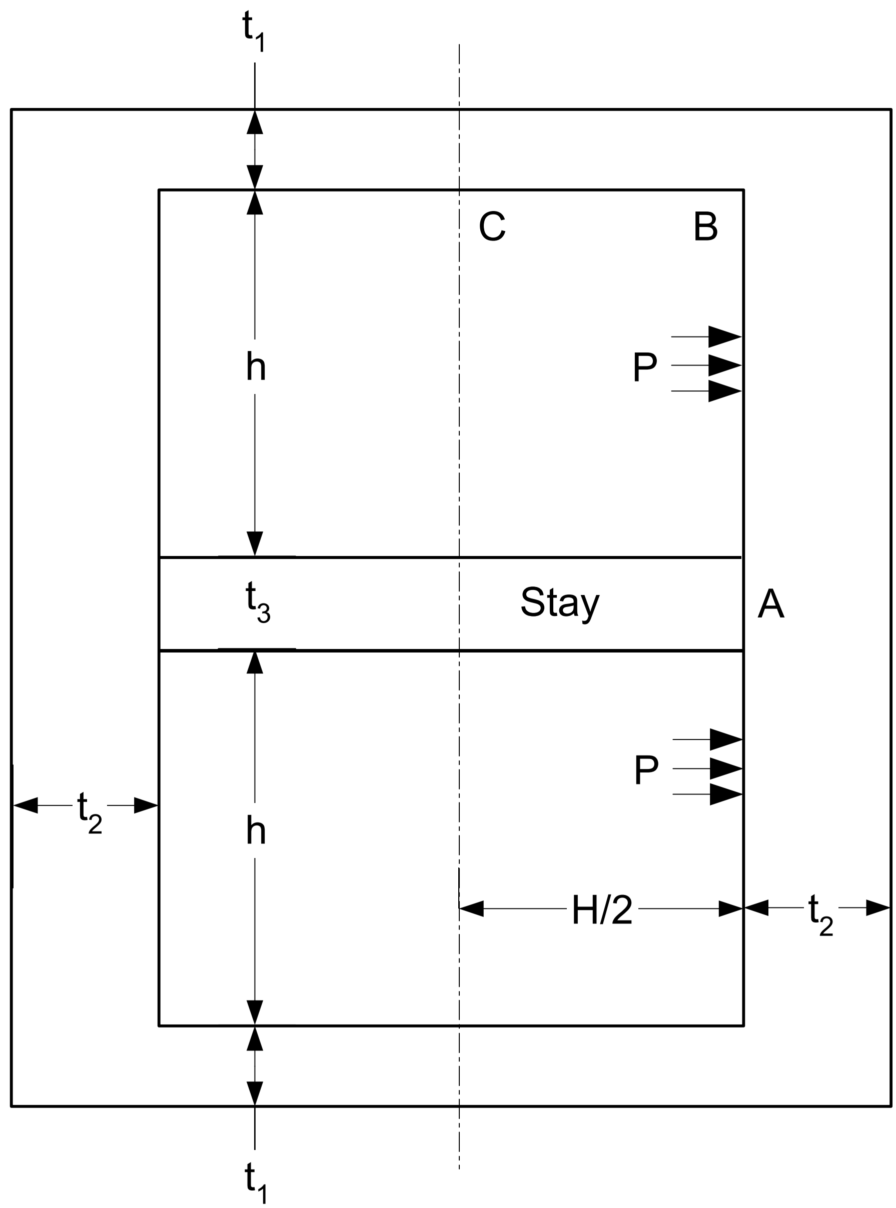
Figure 4.12.6 –Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners)



Notes:

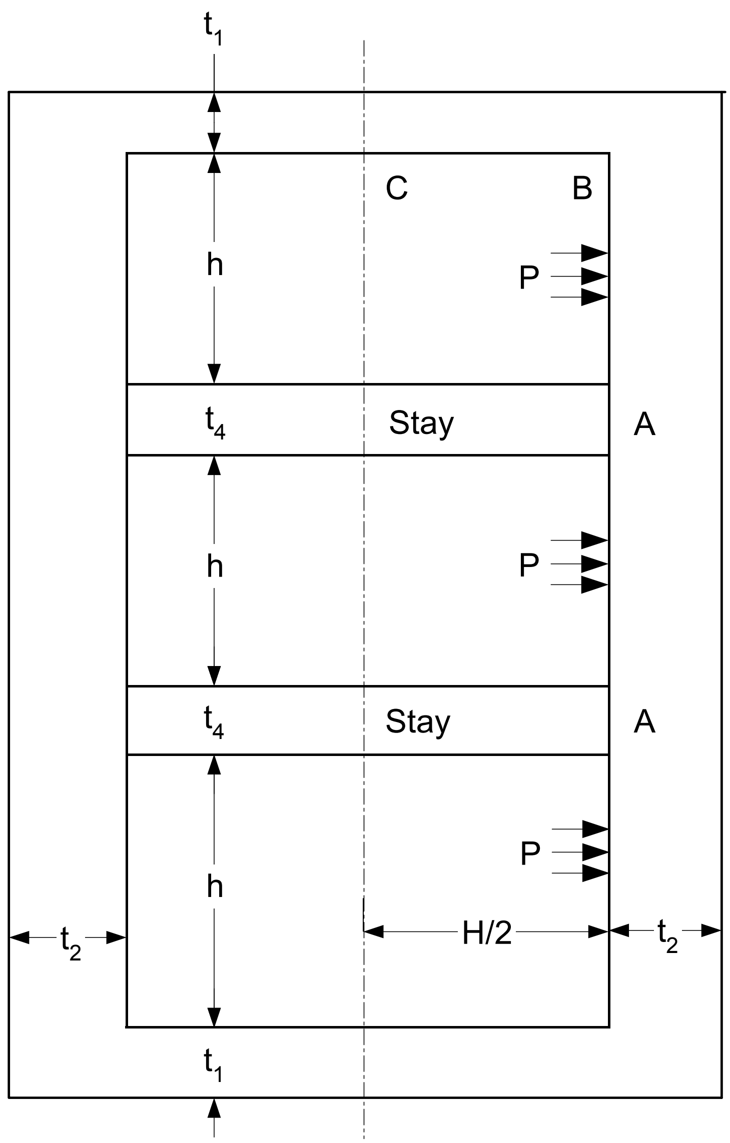
1. Critical Locations of Maximum Stress are defined at points A, B, C, E, F, G, H, M, N, and U.
2. Defined Locations for Stress Calculations are determined using variables X and Y.

Figure 4.12.7 –Type 6 Noncircular Vessels (Reinforced Octagonal Cross Section With Chamfered Corners – Details)



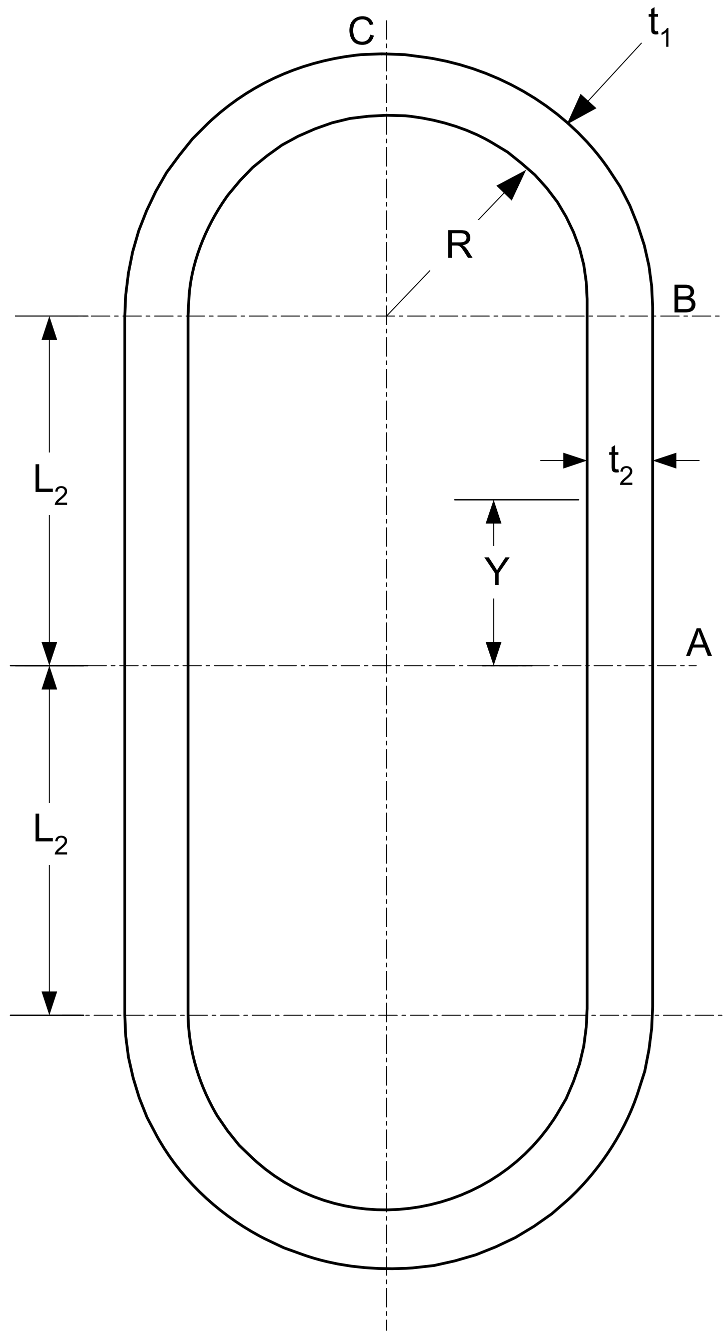
Note: Critical Locations of Maximum Stress are defined at points A, B, and C.

**Figure 4.12.8 – Type 7 Noncircular Vessels
(Rectangular Cross Section With Single Stay Plate Or Multiple Bars)**



Note: Critical Locations of Maximum Stress are defined at points A, B, and C.

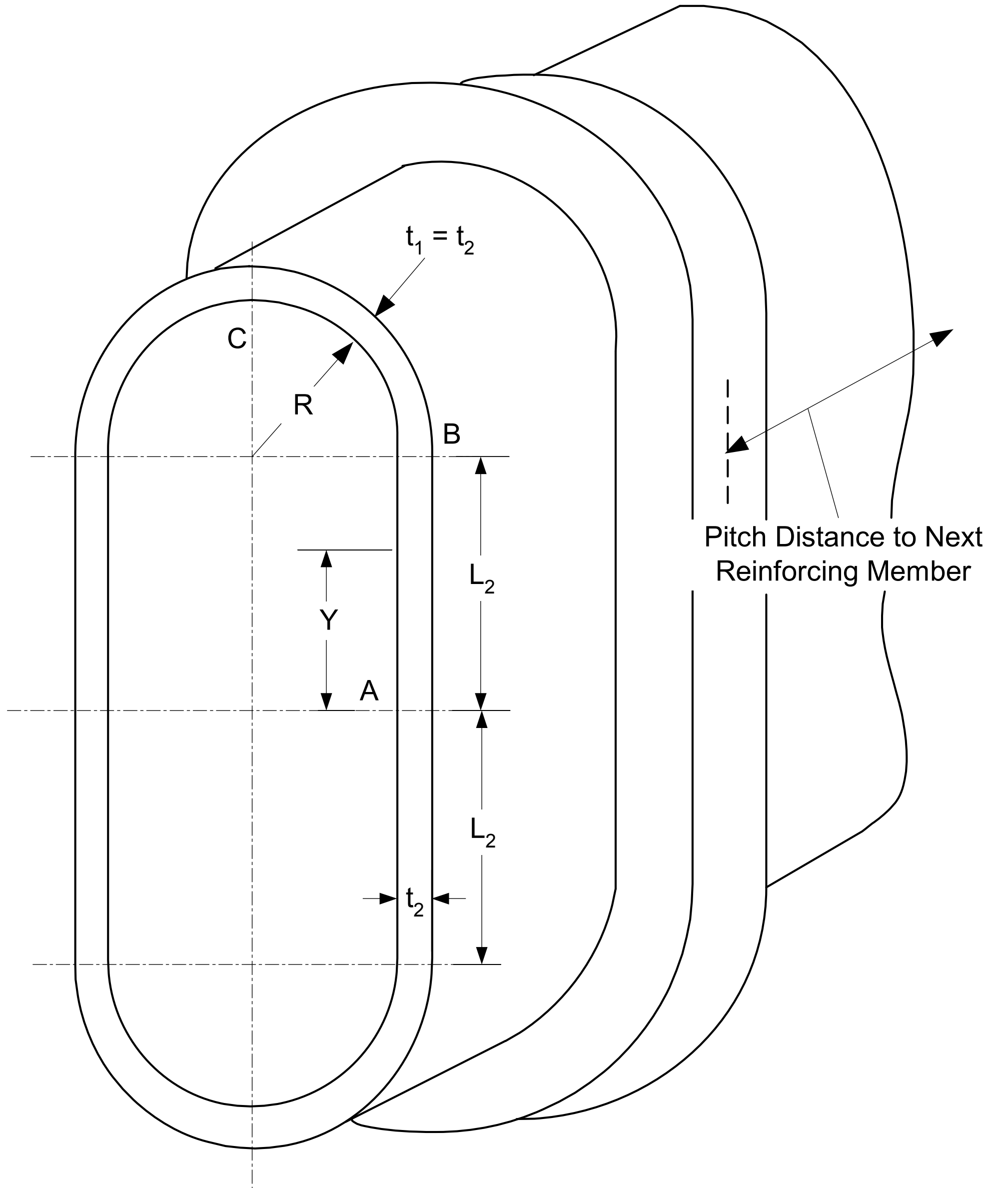
Figure 4.12.9 – Type 8 Noncircular Vessels
(Rectangular Cross Section With Double Stay Plate Or Multiple Bars)



Notes:

1. Critical Locations of Maximum Stress are defined at points A, B, and C.
2. Defined Locations for Stress Calculations are determined using variable Y.

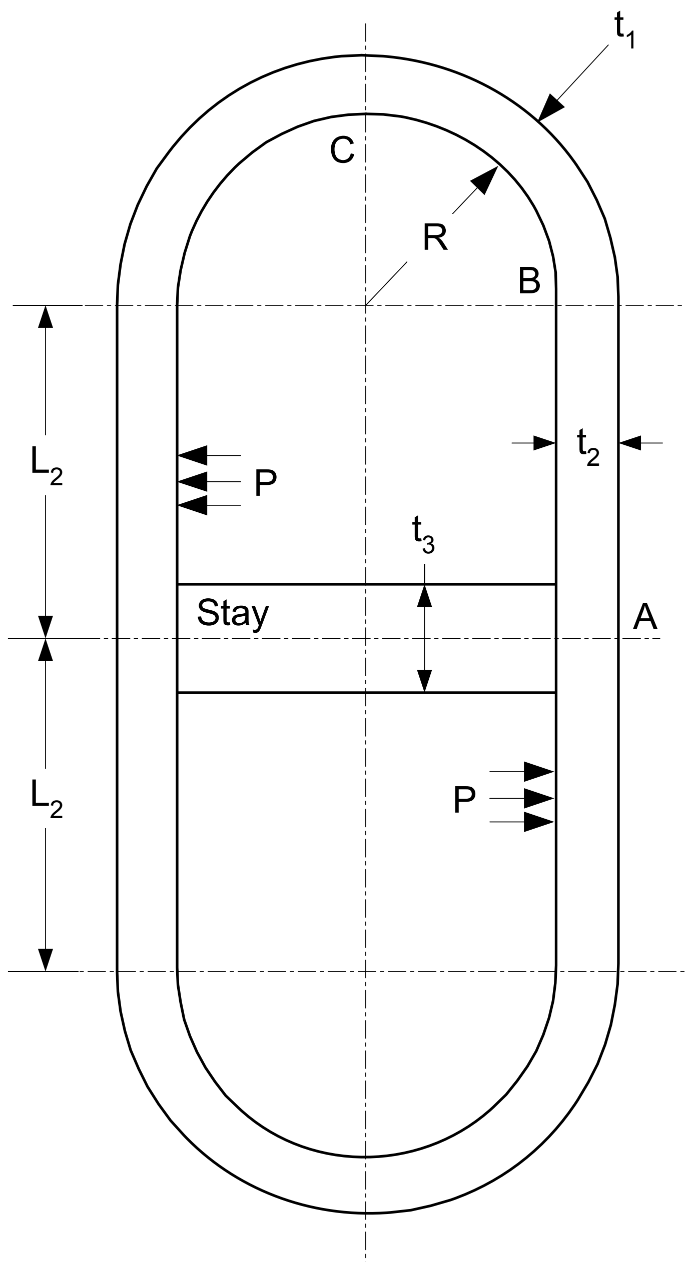
Figure 4.12.10 – Type 9 Noncircular Vessels
(Obround Cross Section)



Notes:

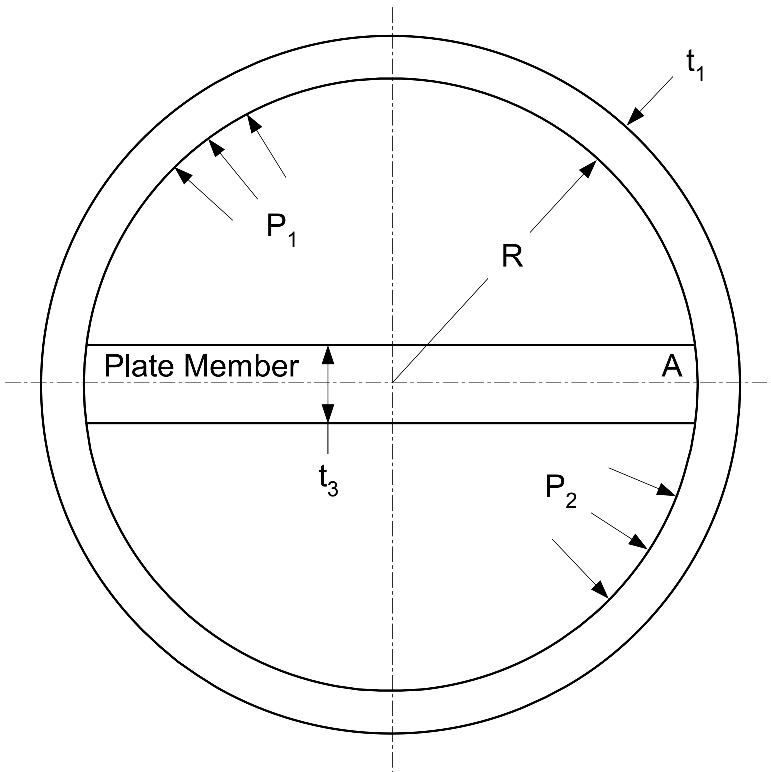
1. Critical Locations of Maximum Stress are defined at points A, B, and C.
2. Defined Locations for Stress Calculations are determined using variable Y.

Figure 4.12.11 – Type 10 Noncircular Vessels (Reinforced Obround Cross Section)



Note: Critical Locations of Maximum Stress are defined at points A, B, and C.

Figure 4.12.12 – Type 11 Noncircular Vessels
(Obround Cross Section With Single Stay Plate Or Multiple Bars)



Note: Critical Locations of Maximum Stress are defined at point A.

Figure 4.12.13 – Type 12 Noncircular Vessels
(Circular Cross Section With Single Stay Plate)

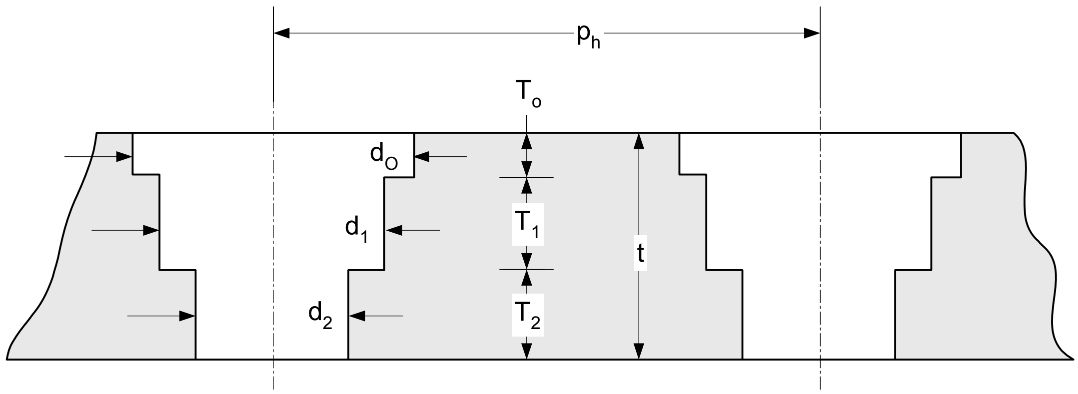
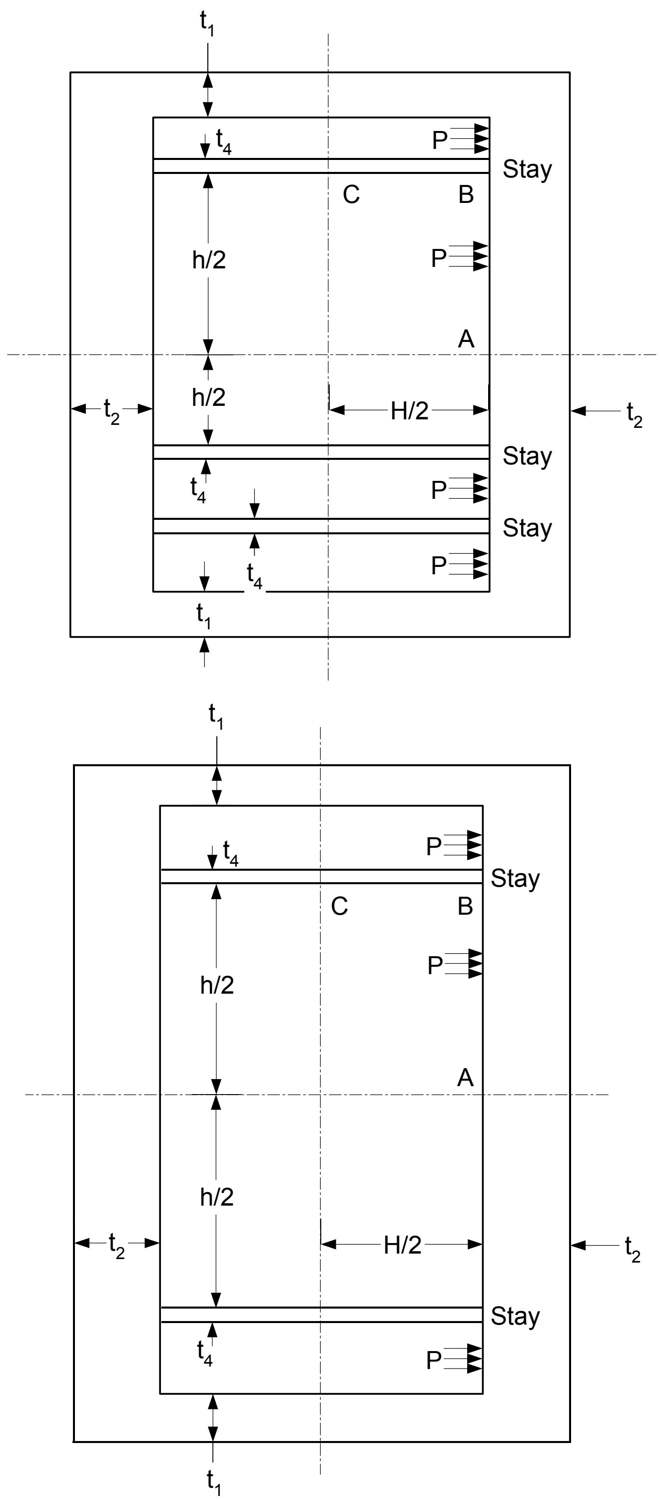


Figure 4.12.14 – Multi-Diameter Holes



Note: Critical Locations of Maximum Stress are defined at points A, B, and C.

Figure 4.12.15 – Rectangular Vessels With Multiple Compartments

4.13 Design Rules for Layered Vessels

4.13.1 Scope

Design rules for layered vessels are covered in paragraph 4.13. There are several manufacturing techniques used to fabricate layered vessels, and these rules have been developed to cover most techniques used today for which there is extensive documented construction and operational data. Examples of acceptable layered shell and head types are shown in Figures 4.13.1 and 4.13.2.

4.13.2 Definitions

The following terms are used in this paragraph to define components of a layered vessel.

a) Layered Vessel – a vessel having a shell and/or heads made up of two or more separate layers.

b) Inner Shell – the inner cylinder that forms the pressure tight membrane.

c) Inner Head – the inner head that forms the pressure tight membrane.

d) Shell Layer – layers may be cylinders formed from plate, sheet, forgings, or the equivalent formed by coiling. This does not include wire winding.

e) Head Layer – anyone of the head layers of a layered vessel except the inner head.

f) Overwraps – layers added to the basic shell or head thickness for the purpose of building up the thickness of a layered vessel for reinforcing shell or head openings, or making a transition to thicker sections of the layered vessel.

g) Dummy Layer – a layer used as a filler between the inner shell (or inner head) and other layers, and not considered as part of the required total thickness.

4.13.3 General

4.13.3.1 The design for layered pressure vessels shall conform to the general design requirements given in paragraph 4.1.

4.13.3.2 A fatigue analysis in accordance with Part 5 shall be performed in all cases unless the fatigue analysis screening based on experience with comparable equipment in accordance with paragraph 5.5.2.2 is satisfied.

4.13.3.3 The Manufacturer's Quality Control System shall include the construction procedure that will outline the sequence and method of application of layers and measurement of layer gaps.

4.13.4 Design for Internal Pressure

4.13.4.1 The total thickness of layered shells of revolution under internal pressure shall not be less than that computed by the equations in paragraph 4.3.

4.13.4.2 An inner shell or inner head material that has a lower allowable design stress than the layer materials may only be included as credit for part of the total wall thickness if S_i is not less than $0.5S_L$ by considering its effective thickness to be:

$$t_{eff} = t_{act}\left(\frac{S_i}{S_L}\right) \tag{4.13.1}$$

4.13.4.3 Layers in which the stress intensity value of the materials is within 20% of the other layers may be used by prorating the allowable stress from Annex 3.A evaluated at the design temperature of the layers in the thickness equation, provided the materials are compatible in modulus of elasticity and coefficient of thermal expansion (see Part 3).

4.13.4.4 The minimum thickness of any layer shall not be less than 3.2 mm (0.125 in.).

4.13.5 Design for External Pressure

4.13.5.1 When layered shells are used for external pressure, the requirements of paragraph 4.4 shall be applied with the following additional requirements.

a) The thickness used for establishing external pressure applied to the outer layer shall be the thickness of the total layers, except as given in paragraph 4.13.5.1.b. The design of the vent holes shall be such that the external pressure is not transmitted through the vent holes in the outer layer.

b) The thickness used for establishing vacuum pressure shall be only the thickness of the inner shell or inner head.

4.13.5.2 Layered shells under axial compression shall be calculated in accordance with paragraph 4.4, utilizing the total layered shell thickness.

4.13.6 Design of Welded Joints

4.13.6.1 The design of welded joints shall conform to the requirements given in paragraph 4.2 except as modified herein.

4.13.6.2 Category A and B joints of inner shells and inner heads of layered sections shall be as follows.

a) Category A joints shall be Type No. 1 (see paragraph 4.2).

b) Category B joints shall be Type No. 1 or Type No. 2 (see paragraph 4.2).

4.13.6.3 Category A joints of layered sections shall be as follows.

a) Category A joints of layers over 22 mm (0.875 in.) in thickness shall be Type No. 1 (see paragraph 4.2).

b) Category A joints of layers 22 mm (0.875 in.) or less in thickness shall be of Type 1 or 2 (see paragraph 4.2), except the final outside weld joint of spiral wrapped layered shells may be a single lap weld.

4.13.6.4 Category B joints of layered shell sections to layered shell sections, or layered shell sections to solid shell sections, shall be of Type 1 or 2 (see paragraph 4.2).

a) Category B joints of layered sections to layered sections of unequal thickness shall have transitions as shown in Figure 4.13.3 Sketch (a) or (b).

b) Category B joints of layered sections to solid sections of unequal thickness shall have transitions as shown in Figure 4.13.3 Sketch (c), (d), (e), or (f).

c) Category B joints of layered sections to layered sections of equal thickness shall be as shown in Figure 4.13.4 Sketch (b), (c), (d), (f), or (g).

d) Category B joints of layered sections to solid sections of equal thickness shall be as shown in Figure 4.13.4 Sketch (a) or (e).

4.13.6.5 Category A joints of solid hemispherical heads to layered shell sections shall be of Type 1 or 2 (see paragraph 4.2).

a) Transitions shall be as shown in Figure 4.13.5 Sketch (a), (b-1), (b-2), or (b-3) when the hemispherical head thickness is less than the thickness of the layered shell section and the transition is made in the layered shell section.

b) Transitions shall be as shown in Figure 4.13.5 Sketch (c), (d-1), or (e) when the hemispherical head thickness is greater than the thickness of the layered shell section and the transition is made in the layered shell section.

c) Transition shall be as shown in Figure 4.13.5 Sketch (f) when the hemispherical head thickness is less than the thickness of the layered shell section and the transition is made in the hemispherical head section.

4.13.6.6 Category B joints of solid elliptical, torispherical, or conical heads to layered shell sections shall be of Type 1 or 2 (see paragraph 4.2). Transitions shall be as shown in Figure 4.13.5 Sketch (c), (d-1), (d-2), (e), or (f).

4.13.6.7 Category C joints of solid flat heads and tube-sheets to layered shell sections shall be of Type 1 or 2 (see paragraph 4.2) as indicated in Figure 4.13.6. Transitions, if applicable, shall be used as shown in Figure 4.13.3 Sketch (c), (d), (e), or (f).

4.13.6.8 Category C joints attaching solid flanges to layered shell sections and layered flanges to layered shell sections shall be of Type 1 or 2 (see paragraph 4.2) as indicated in Figure 4.13.7.

4.13.6.9 Category A joints of layered hemispherical heads to layered shell sections shall be of Type 1 or 2 (see paragraph 4.2) with transition as shown in Figure 4.13.8 Sketch (a-1) or (a-2).

4.13.6.10 Category B joints of layered conical heads to layered shell sections shall be of Type 1 or 2 (see paragraph 4.2) with transitions as shown in Figure 4.13.8 Sketch (b-1).

4.13.6.11 Category B joints of layered shell sections to layered shell sections or layered shells to solid heads or shells may be butt joints as shown in Figure 4.13.4 Sketches (c), (d), and (e), or step welds as shown in Figure 4.13.4 Sketches (a), (b), (f), and (g).

4.13.6.12 Category D joints of solid nozzles, manholes, and other connections to layered shell or layered head sections shall be full penetration welds as shown in Figure 4.13.9 except as permitted in Sketch (i), (j), (k), or (l). Category D joints between layered nozzles and shells or heads are not permitted.

4.13.6.13 When layers of Category A joints as shown in Figure 4.13.5 Sketches (a), (b-1), (b-2), and (b-3) and Figure 4.13.8 Sketches (a-1) and (a-2) are welded with fillet welds having a taper less than 3:1, an analysis of the head-to-shell junction shall be done in accordance with Part 5. Resistance due to friction shall not be considered in the analysis. The longitudinal load resisted by the weld shall consider the load transferred from the remaining outer layers.

4.13.7 Nozzles and Nozzle Reinforcement

4.13.7.1 All openings, except as provided in paragraph 4.13.7.2 shall meet the requirements for reinforcing per paragraph 4.5. All reinforcements required for openings shall be integral with the nozzle or provided in the layered section or both. Additional layers may be included for required reinforcement.

4.13.7.2 Openings, DN 50 (NPS 2) and smaller, need not be reinforced when installed in layered construction but may be welded on the inside as shown in Figure 4.13.9 Sketch (j). The nozzle nominal wall thickness shall not be less than Schedule 80 pipe as fabricated in addition to meeting the requirements of paragraph 4.5.5.

4.13.7.3 Some acceptable nozzle geometries and attachments are shown in Figure 4.13.9.

4.13.7.4 Openings up to and including 6 in. nominal pipe size (DN 150) may be constructed as shown in Figure 4.13.9 Sketches (k) and (l). Such partial penetration weld attachments may only be used for instrumentation openings, inspection openings, etc. on which there are no external mechanical loadings provided the following requirements are met.

a) The requirements for reinforcing specified in paragraph 4.13.7 apply except that the diameter of the finished openings in the wall shall be d^* as specified in Figure 4.13.9 Sketches (k) and (l), and the thickness t_r is the required thickness of the layered shells computed by the design requirements.

b) Additional reinforcement, attached to the inside surface of the inner shell, may be included after the corrosion allowance is deducted from all exposed surfaces. The attachment welds shall comply with paragraph 4.2 and Figure 4.13.9 Sketch (k) or (l).

c) Metal in the nozzle neck available for reinforcement shall be limited by the boundaries specified in paragraph 4.5 except that the inner layer shall be considered the shell.

4.13.7.5 Openings greater than 51 mm (2 in.) may be constructed as shown in Figure 4.13.9 Sketch (i). The requirements for reinforcing specified in paragraph 4.13.7.4.a apply except that:

a) The diameter of the finished openings in the walls shall be d' as specified in Figure 4.13.9 Sketch (i), and the thickness t_r is the required thickness of the layered shells computed by the design requirements;

b) Additional reinforcement may be included in the solid hub section as shown in Figure 4.13.9 Sketch (i);

c) Metal in the nozzle neck available for reinforcement shall be limited by the boundaries specified in paragraph 4.5, except that the inner layer shall be considered the shell.

4.13.8 Flat Heads

4.13.8.1 Design criteria shall meet the requirements of paragraph 4.6.

4.13.8.2 The design of welded joints shall be in accordance with paragraph 4.13.6.

4.13.9 Bolted and Studded Connections

4.13.9.1 Design criteria shall meet the requirements of paragraph 4.16.

4.13.9.2 The design of welded joints shall be in accordance with paragraph 4.13.6.

4.13.10 Attachments and Supports

4.13.10.1 Supports for layered pressure vessels may be designed in accordance with paragraph 4.15. Examples of some acceptable supports are shown in Figure 4.13.10.

4.13.10.2 When attaching supports or other connections to the outside or inside of layered pressure vessels, only the immediate layer shall be used in the calculation, except where provisions are made to transfer the load to other layers.

4.13.10.3 When jacketed closures are used, provisions shall be made for extending layer vents through the jacket (see paragraph 4.13.11.1). Partial jackets covering only a portion of the circumference are not permitted on layered shells.

4.13.11 Vent Holes

4.13.11.1 Vent holes shall be provided to detect leakage of the inner shell and to prevent buildup of pressure within the layers as follows.

4.13.11.2 In each shell course or head segment, a layer may be made up of one or more plates. Each layer plate shall have at least two vent holes 6 mm (0.25 in.) minimum diameter. Holes may be drilled radially through the multiple layers or may be staggered in individual layer plates.

4.13.11.3 For continuous coil wrapped layers, each layered section shall have at least four vent holes 6 mm (0.25 in.) minimum diameter. Two of these vent holes shall be located near each end of the section and spaced approximately 180 deg apart.

4.13.11.4 The minimum requirement for spirally wound strip layered construction shall be 6 mm (0.25 in.) minimum diameter vent holes drilled near both edges of the strip. They shall be spaced for the full length of the strip and shall be located a distance of approximately $\pi R / \tan\theta$ from each other (where R is the mean radius of the shell and θ is the acute angle of spiral wrap measured from the longitudinal centerline, deg).

4.13.11.5 If a strip weld covers a vent hole, partially or totally, an additional vent hole shall be drilled on each side of the obstructed hole.

4.13.11.6 In addition to the above, holes may be drilled radially through the multiple layers.

4.13.11.7 Vent holes shall not be obstructed. If a monitoring system is used, it shall be designed to prevent buildup of pressure within the layers.

4.13.12 Shell Tolerances

4.13.12.1 Contact Between Layers

The following requirements shall be satisfied.

a) Category A weld joints shall be ground to ensure contact between the weld area and the succeeding layer, before application of the layer.

b) Category A weld joints of layered shell sections shall be in an offset pattern so that the centers of the welded longitudinal joints of adjacent layers are separated circumferentially by a distance of at least five times the layer thickness.

c) Category A weld joints in layered heads may be in an offset pattern; if offset, the joints of adjacent layers shall be separated by a distance of at least five times the layer thickness.

d) After weld preparation and before welding circumferential seams, the height of the radial gaps between any two adjacent layers shall be measured at the ends of the layered shell section or layered head section at right angles to the vessel axis, and also the length of the relevant radial gap in inches shall be measured (neglecting radial gaps of less than 0.25 mm (0.010 in.) as non relevant). The gap area, A_g, shall not exceed the thickness of a layer expressed in square inches. An approximation of the area of the gap shall be calculated using Equation (4.13.2). The maximum length of any gap shall not exceed the inside diameter of the vessel. Where more than one gap exists between any two adjacent layers, the sum of the gap lengths shall not exceed the inside diameter of the vessel. The maximum height of any gap shall not exceed 4.8 mm (0.1875 in.). It is recognized that there may be vessels of dimensions wherein it would be desirable to calculate a maximum permissible gap area, and also when cyclical service conditions require it. This procedure is provided in paragraph 4.13.12.2 and may be used in lieu of the maximum gap area given above, (see Figure 4.13.11).

$$A_g = \frac{2}{3}bh \tag{4.13.2}$$

e) In the case of layered spheres or layered heads, if the gaps cannot be measured as required in paragraph 4.13.12.1.d, measurement of gap heights shall be taken through vent holes in each layer course to assure that the height of layer gaps between any two layers does not exceed the gap permitted in paragraph 4.13.12.1.d. The spacing of the vent holes shall be such that gap lengths can be determined. In the event an excessive gap height is measured through a vent hole, additional vent holes shall be drilled as required to determine the gap length. There shall be at least one vent hole per layer segment.

4.13.12.2 Alternative to Measuring Contact between Layers During Construction

As an alternative to paragraph 4.13.12.1.d, the following measurements shall be taken at the time of the hydrostatic test to check on the contact between successive layers, and the effect of gaps which may or may not be present between layers.

a) The circumference shall be measured at the midpoint between adjacent circumferential joints, or between a circumferential joint and any nozzle in a shell course. Measurements shall be taken at zero pressure and, following application of hydrostatic test pressure, at the design pressure. The difference in measurements shall be averaged for each course in the vessel and the results recorded as average middle circumferential expansion, e_m.

b) The theoretical circumferential expansion of a solid vessel of the same dimensions and materials as the layered vessel shall be calculated from Equation (4.13.3). The acceptance criterion for circumferential expansion at the design pressure is: $e_m \geq 0.5e_{th}$.

$$e_{th} = \frac{1.7\pi P\left(2R_m - t_s\right)^2\left(2R_m + t_s\right)}{8E_y R_m t_s} \tag{4.13.3}$$

4.13.12.3 Rules for Calculating Maximum Permissible Gaps

The maximum number and size of gaps permitted in any cross section of a layered vessel shall be limited by paragraphs 4.13.12.3.a and 4.13.12.3.b.

a) Maximum gap between any two layers shall not exceed the value of h given by Equation (4.13.4):

$$h = 0.55\left(N - 0.5 - \frac{P}{S_m}\right)\frac{R_g S_m}{E_y} \tag{4.13.4}$$

where,

$$N = 3 \qquad \textit{for infinite cycles} \tag{4.13.5}$$

$$N = \frac{2}{K_c}\left(\frac{S_a}{S_m}\right) \qquad \textit{for a specified number of cycles} \tag{4.13.6}$$

with,

$$K_c = \sqrt{\frac{4S_a}{3S_m} + 0.25 - 0.5} \tag{4.13.7}$$

b) Maximum permissible number of gaps and their corresponding arc lengths at any cross section of a layered vessel shall be calculated as follows.

1) Measure each gap and its corresponding length throughout the cross section.

2) Calculate the value of F for each of the gaps using the following equation:

$$F = 0.109\left(\frac{bh}{R_g^2}\right) \tag{4.13.8}$$

3) The total sum of the calculated F values shall not exceed the quantity

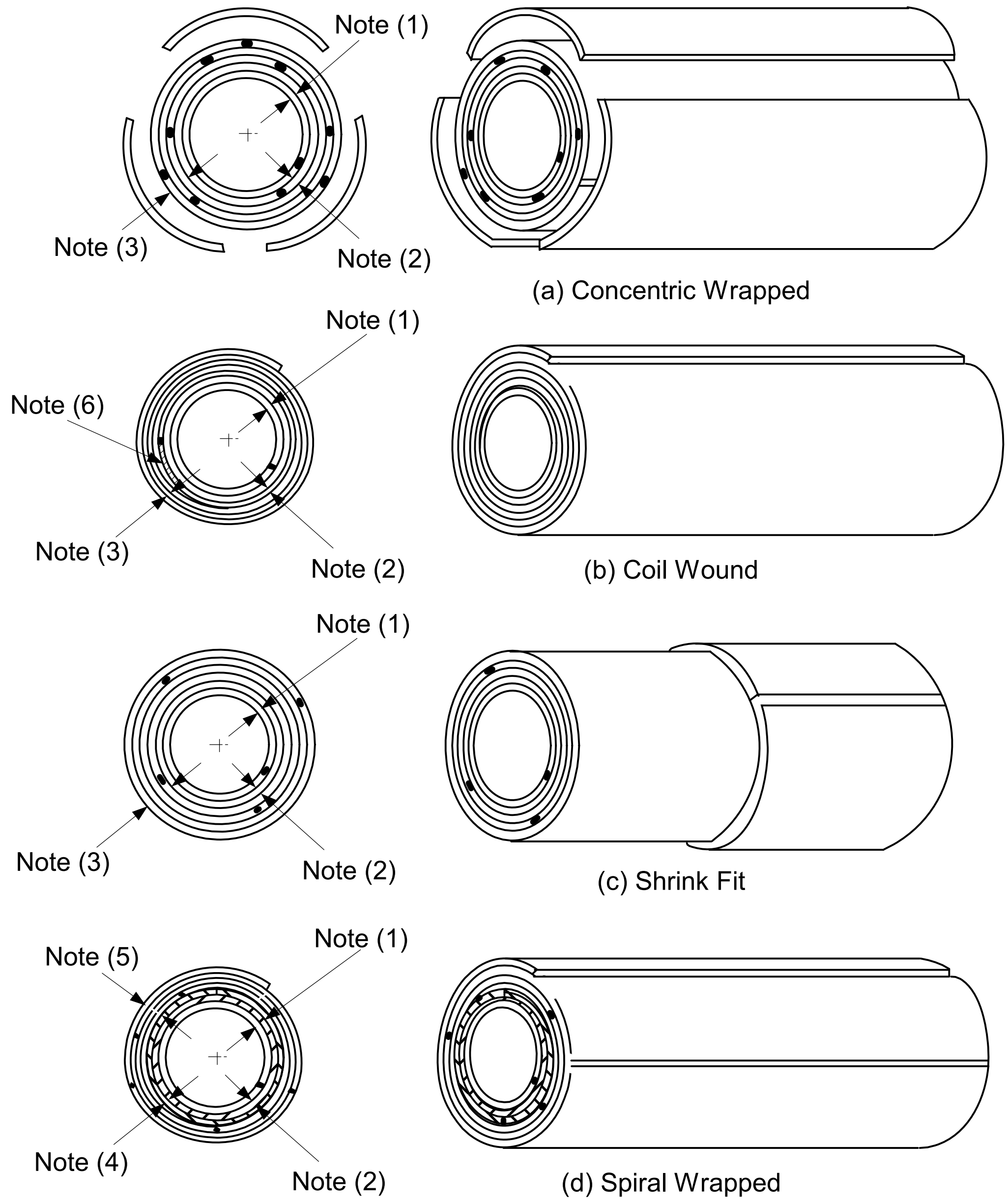
$$F_T = \frac{1-\nu^2}{E_y}\left(NS_m - \frac{2PR_o^2}{R_o^2 - R_i^2}\right) \tag{4.13.9}$$

4.13.13 Nomenclature

A_g	gap area.
b	length of the gap between any two layers.
C	equal to 3 mm (0.125 in.) radial clearance between the nozzle neck and vessel opening
d^*	finished opening in the wall
e_{th}	theoretical circumferential expansion.
e_m	average middle recorded circumferential expansion.
F	gap value.
F_T	total permissible gap value.
E_y	Modulus of Elasticity for the layer material from Part 3 .
h	gap between any two layers.
ν	Poisson's ratio.
P	design pressure of the vessel.
R_g	outside radius of the layer above where the gap is located.
R_i	inside radius of the vessel.
R_m	mean radius of the vessel.
R_o	outside radius of the vessel.
S_a	stress amplitude from the applicable fatigue curve for the layer material from Annex 3.F .
S_L	allowable stress for the layers from Annex 3.A at the design temperature.
S_i	allowable stress for the inner layer from Annex 3.A at the design temperature.
S_m	allowable stress for the layer material from Annex 3.A at the design temperature.
r_1	equal to $\min\left[0.25t_n, 3mm\left(0.125in\right)\right]$
r_2	equal to 6 mm (0.25 in.) minimum
r_3	equal to $\min\left[0.25t_n, 19mm\left(0.75in\right)\right]$
t	actual thickness of the head or tubesheet or for nozzle details equal to $\min\left[t_n,\ 19\ mm\ (0.75 in.)\right]$, as applicable.

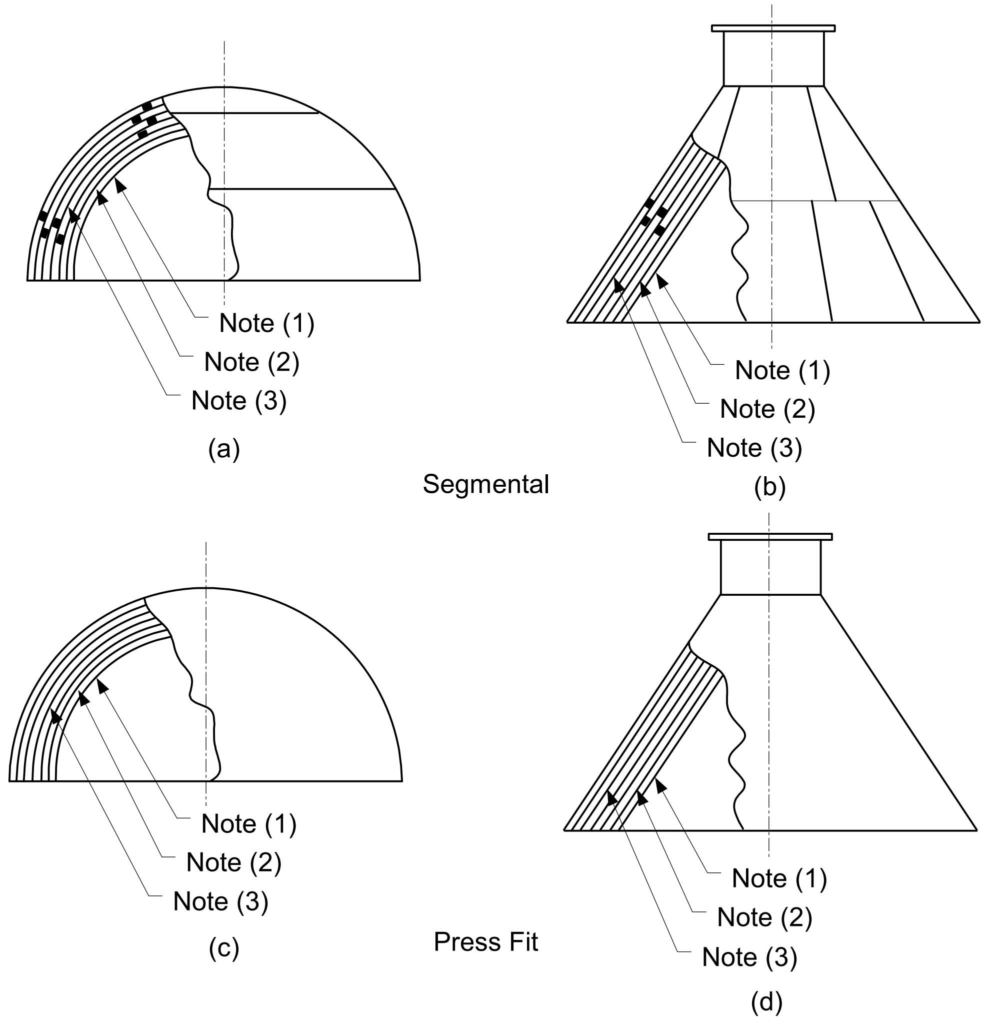
t_{act}	actual thickness of inner shell or inner head.
t_c	equal to the larger of 6 mm (0.25 in.) or $0.7\min\left[t_n, 19\ mm\ (0.75\,in.)\right]$
t_{eff}	effective thickness of inner shell or inner head.
t_H	thickness of the head at the head-to-cylinder joint.
t_L	thickness of the layer.
t_n	nominal thickness of the nozzle wall less corrosion allowance
t_S	total wall thickness of the layered vessel.
Y	offset.

4.13.14 Figures



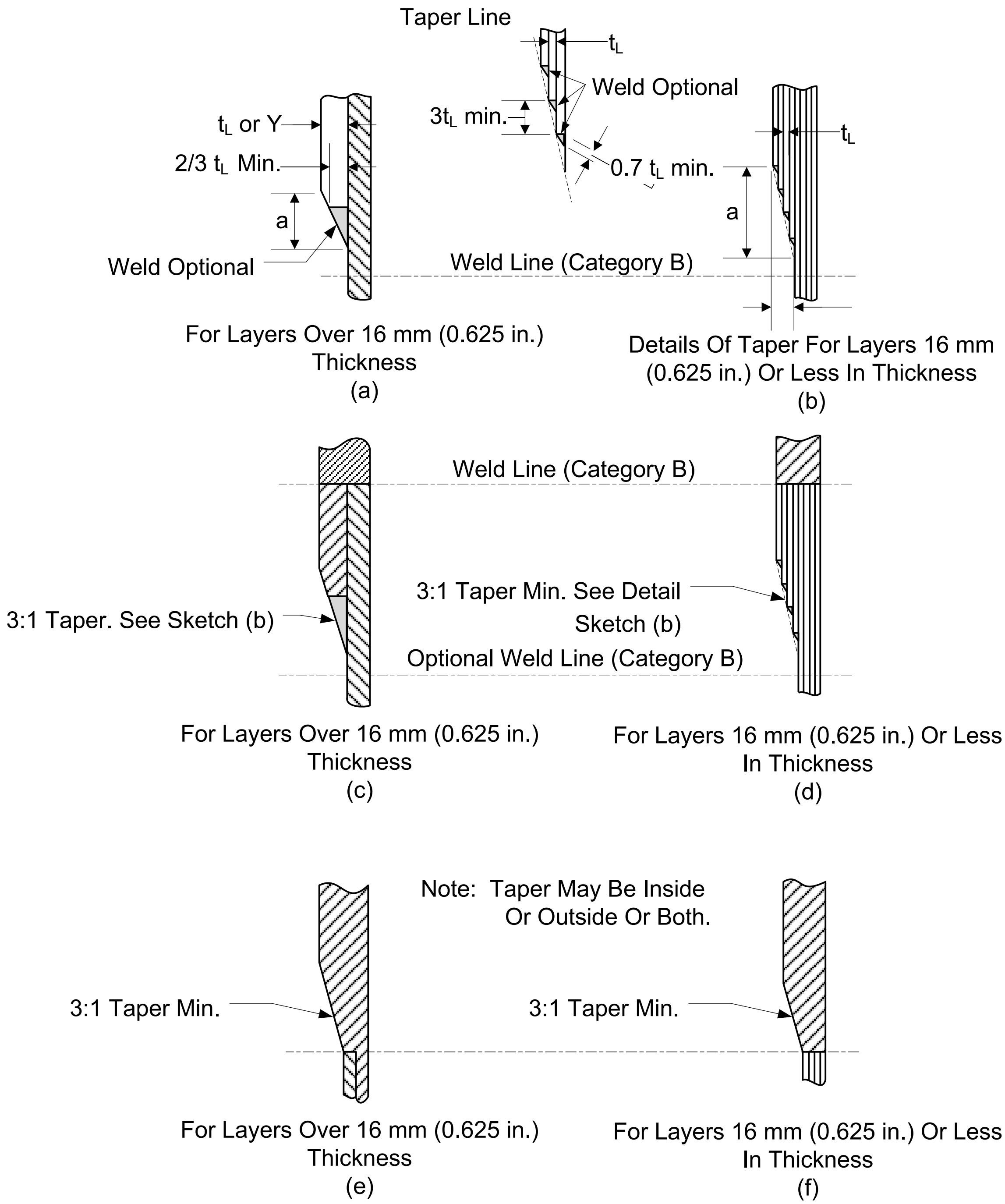
Notes:
1. Inner shell
2. Dummy layer (if used)
3. Layers
4. Shell layer (tapered)
5. Balance of layers
6. Gap

Figure 4.13.1 – Some Acceptable Layered Shell Types



Notes:
1. Inner head
2. Dummy layer (if used)
3. Head layers

Figure 4.13.2 – Some Acceptable Layered Head Types



Notes:
1. $a \geq 3Y$ where a is the required length of the taper and Y is the offset.
2. The length of the required taper may include the width of the weld.
3. The transition may be on either or both sides.

Figure 4.13.3 – Transitions of Layered Shell Sections

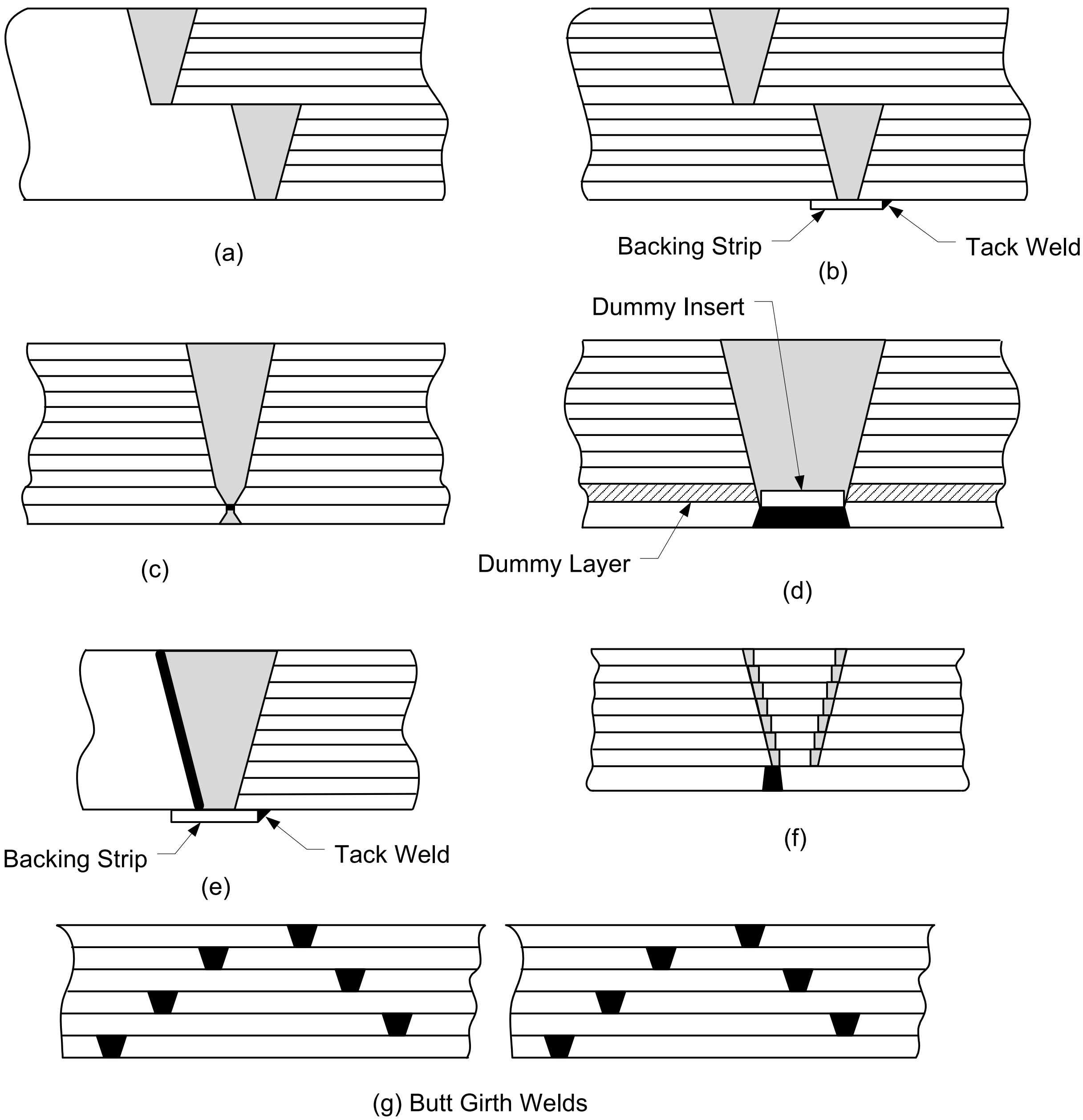


Figure 4.13.4 – Some Acceptable Welded Joints of Layered-To-Layered and Layered-To-Solid Sections

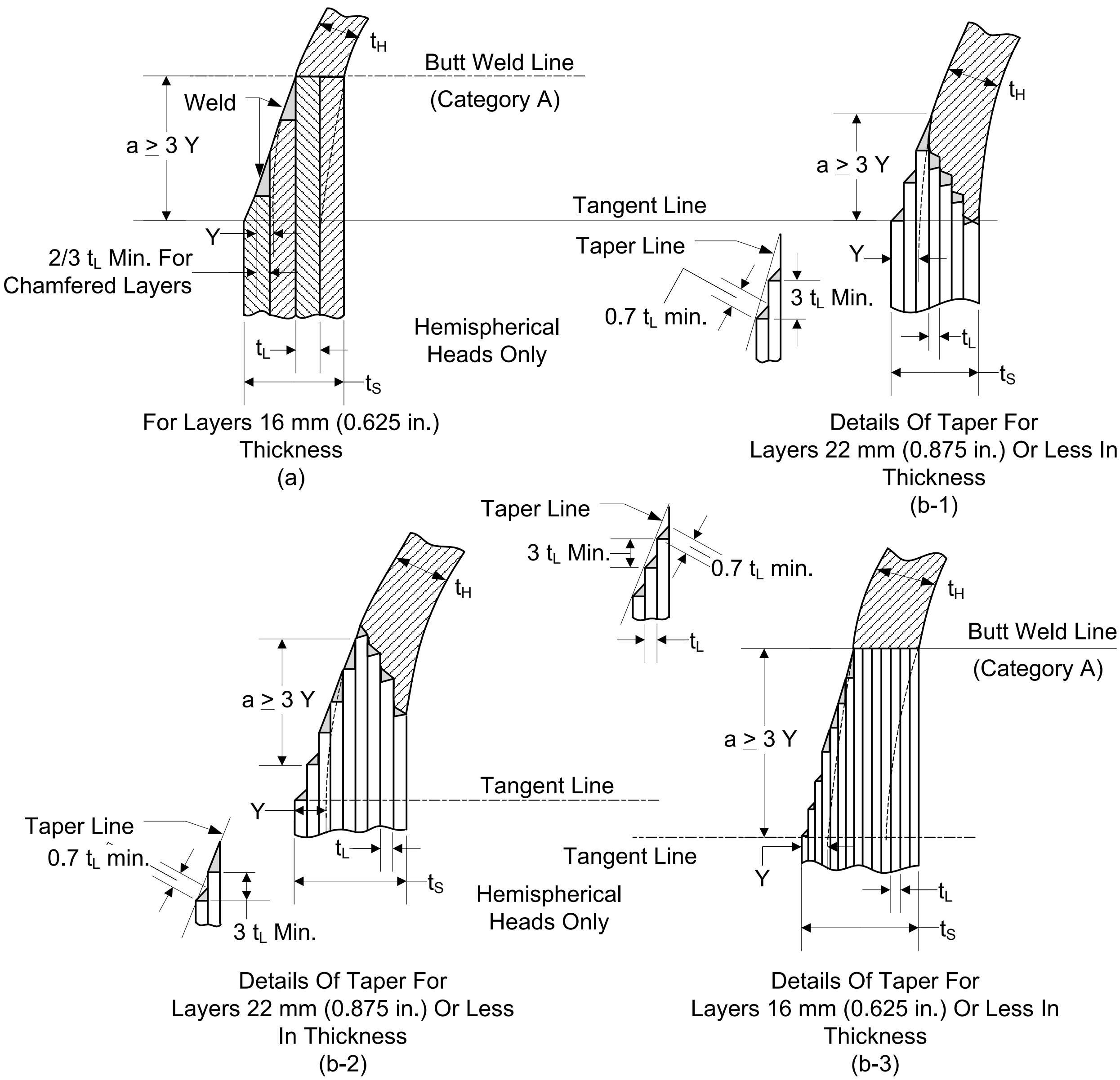
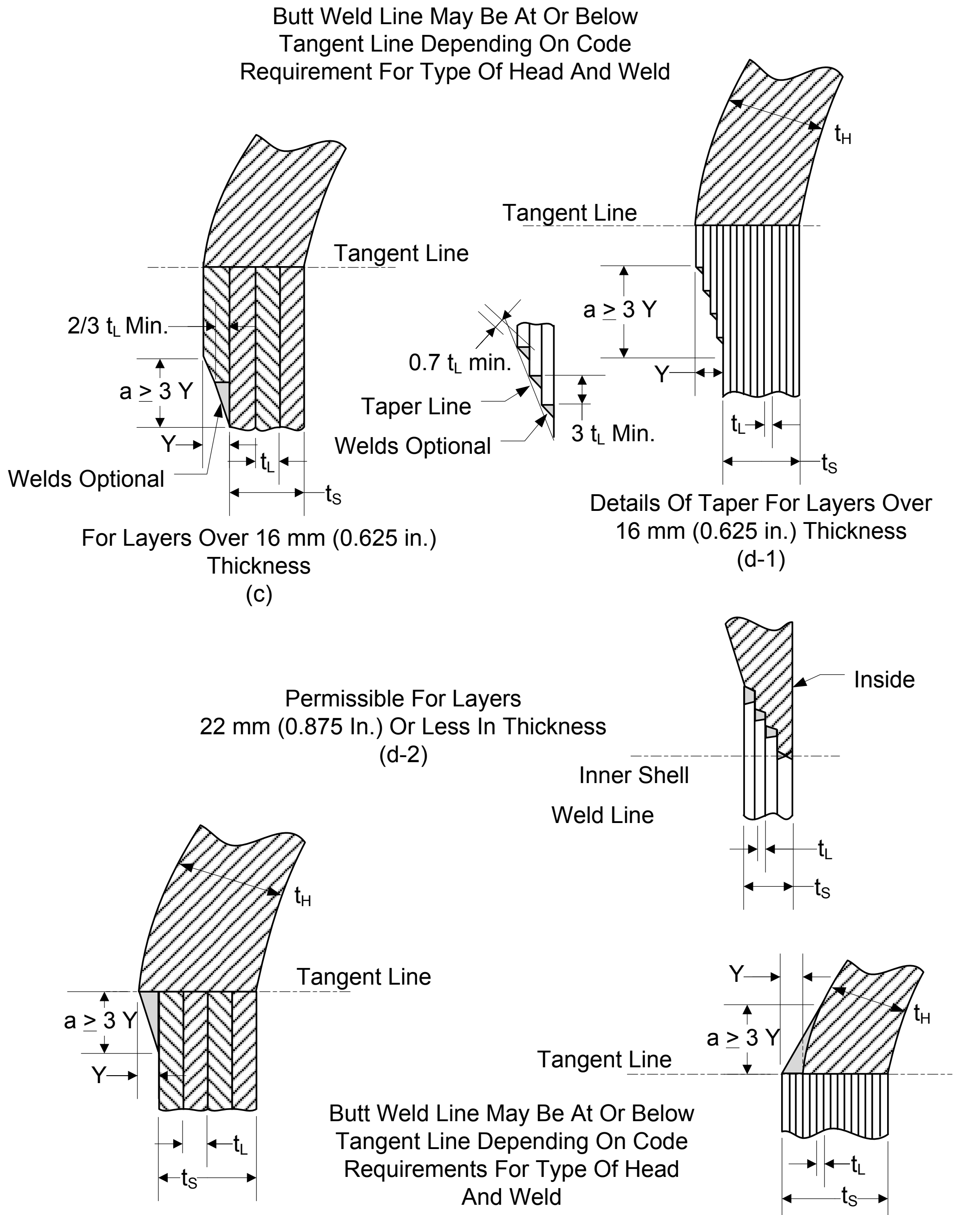


Figure 4.13.5 – Some Acceptable Solid Head Attachments to Layered Shell Sections



For Layers Over 16 mm (0.625 in.) Thickness
(c)

Details Of Taper For Layers Over 16 mm (0.625 in.) Thickness
(d-1)

Permissible For Layers 22 mm (0.875 In.) Or Less In Thickness
(d-2)

For Layers 16 mm (0.625 in.) Or Less In Thickness
(e)

For Layers Of Any Thickness
(f)

Notes:

1. In Sketch (e), $Y \le t_L$ shall be satisfied, in Sketch (f), $Y \le 0.5 t_S$ shall be satisfied
2. In all cases, $a \ge 3Y$ shall be satisfied. The shell centerline may be on either side of the head centerline by a maximum distance of $0.5(t_S - t_H)$. The length of the required taper may include the width of the weld.
3. The actual thickness shall not be less than the theoretical head thickness.

Figure 4.13.5 – Some Acceptable Solid Head Attachments to Layered Shell Sections (Continued)

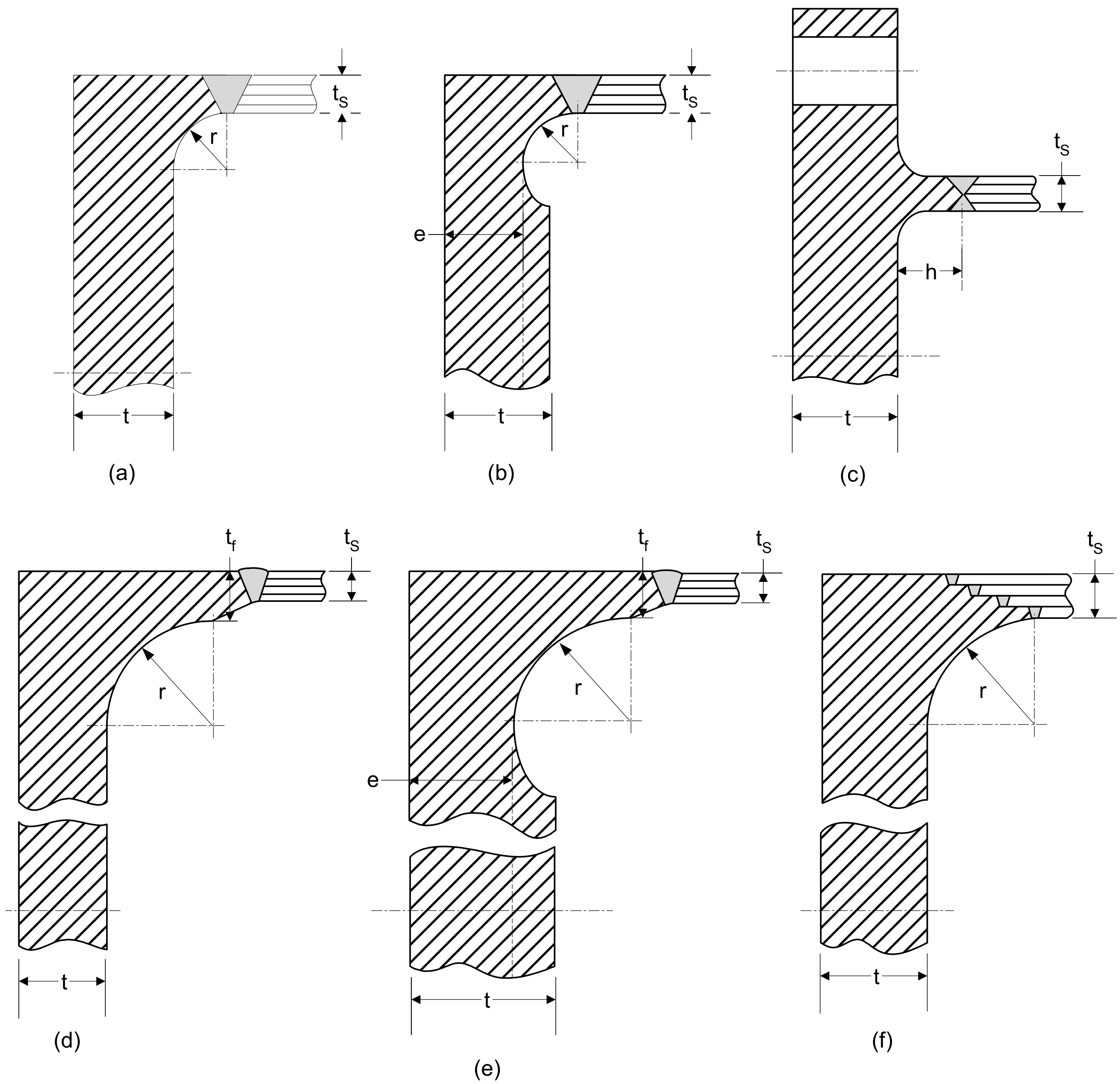
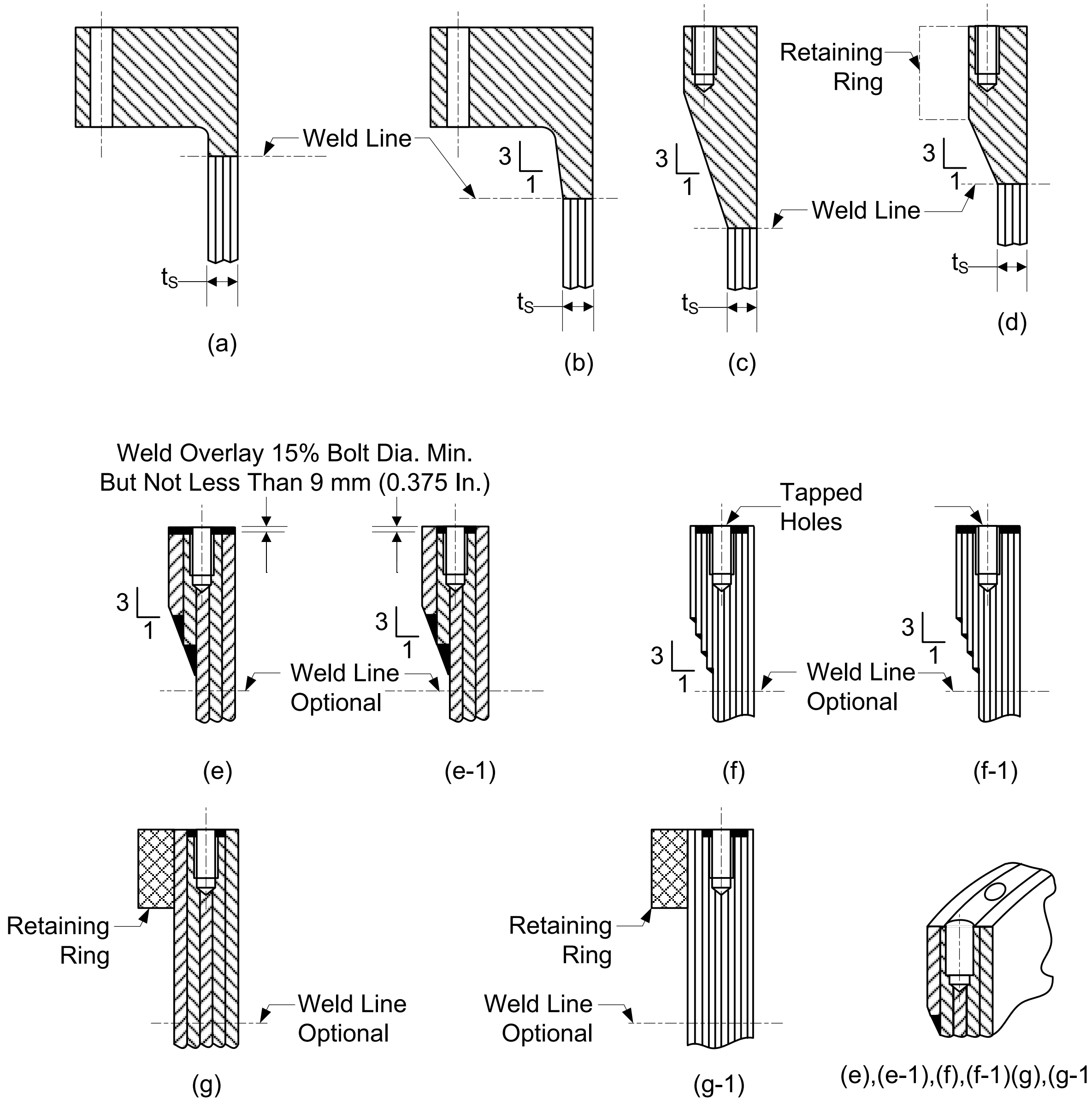


Figure 4.13.6 – Some Acceptable Flat Heads and Tubesheets With Hubs Joining Layered Shell Sections



Notes:

1. The following applies to Sketches (e), (e-1), (f), (f-1), (g), and (g-1): the weld overlay shall tie the overlay, the overwraps, and layers together, and the bolt circle shall not exceed the outside diameter of the shell.
2. For Sketches (e), (e-1), (f), and (f-1), the angle of the transition and size of the fillet welds are optional, and the bolt circle shall not exceed the outside diameter of the shell.

Figure 4.13.7 – Some Acceptable Flanges for Layered Shells

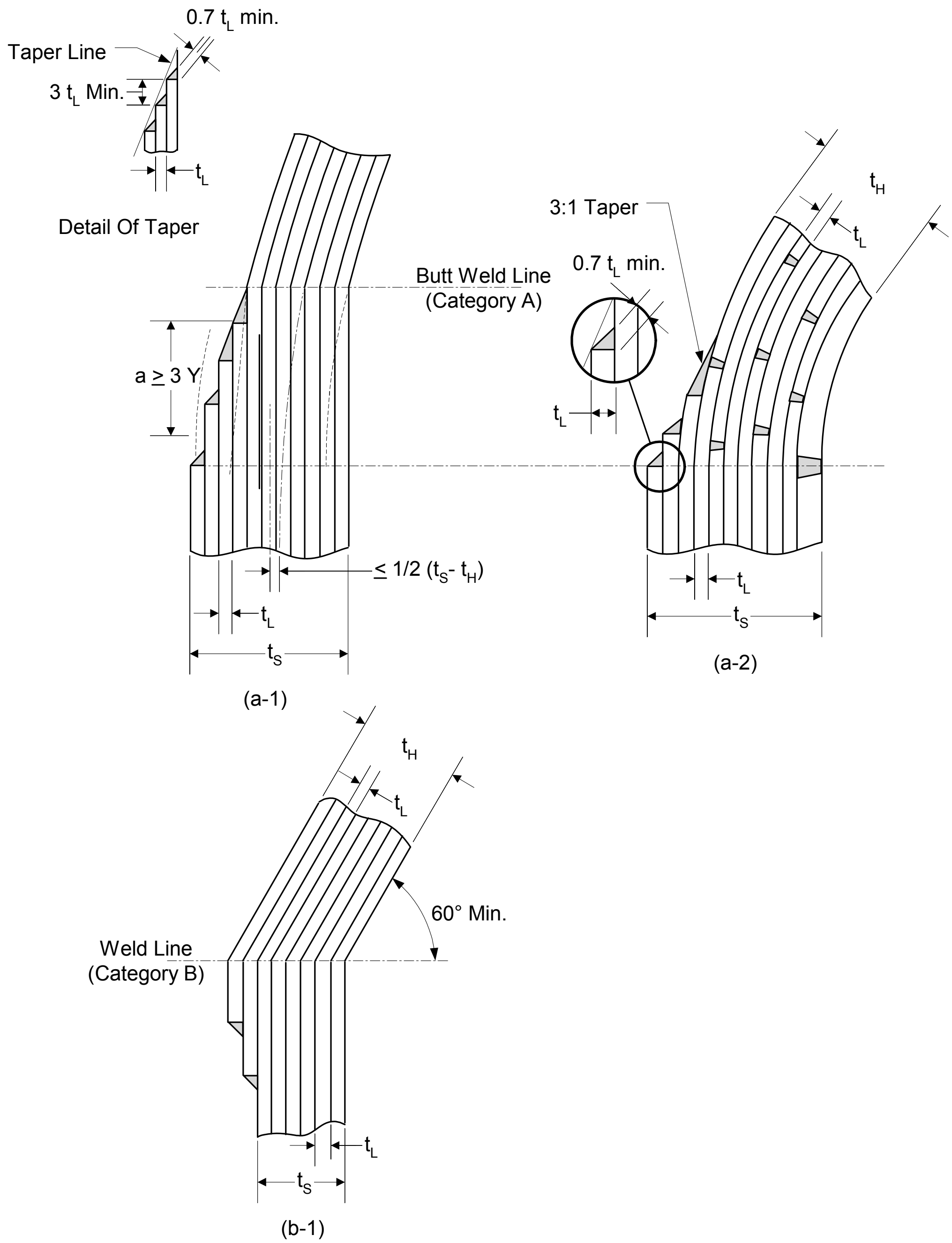


Figure 4.13.8 – Some Acceptable Layered Head Attachments to Layered Shells

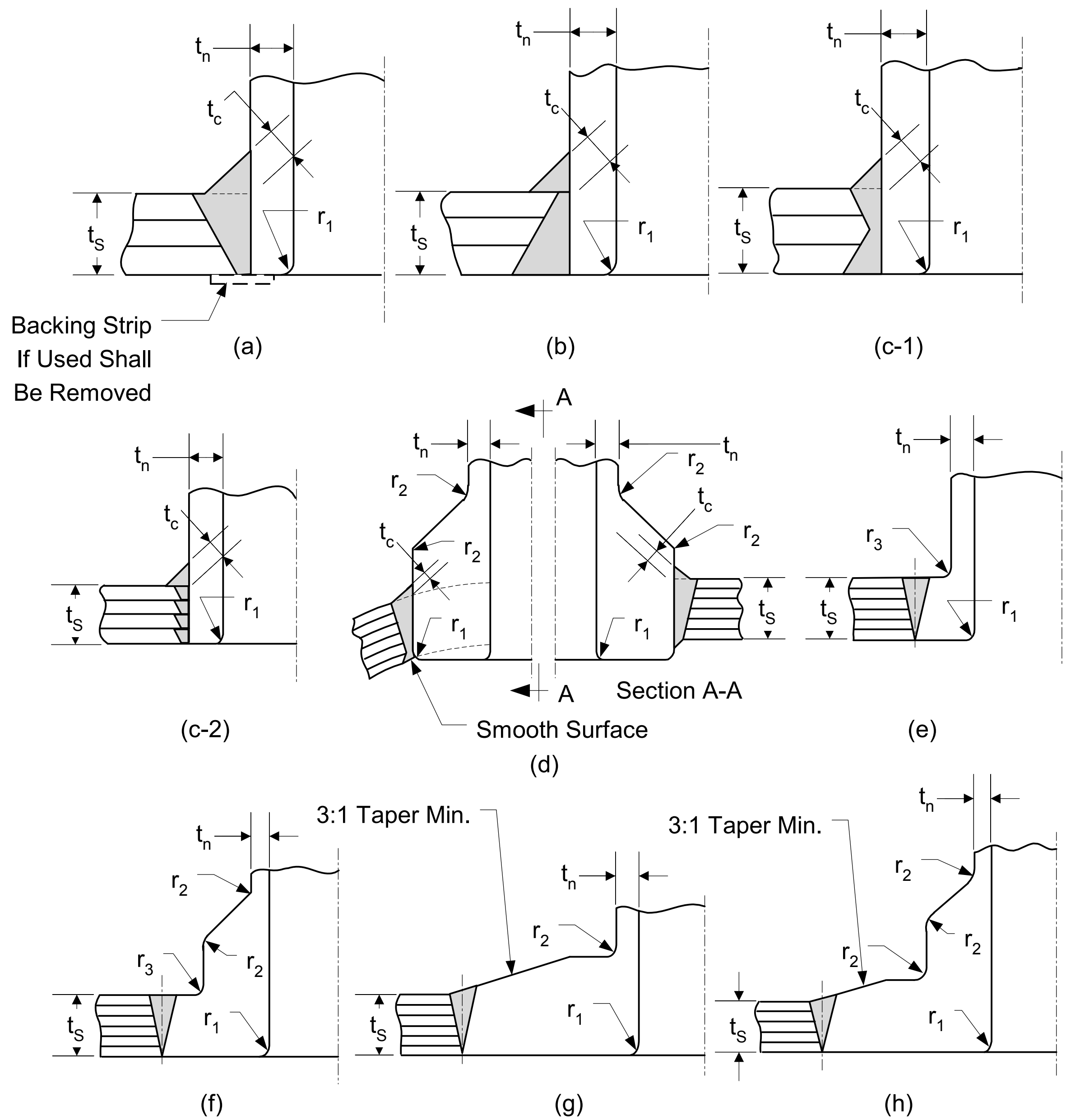
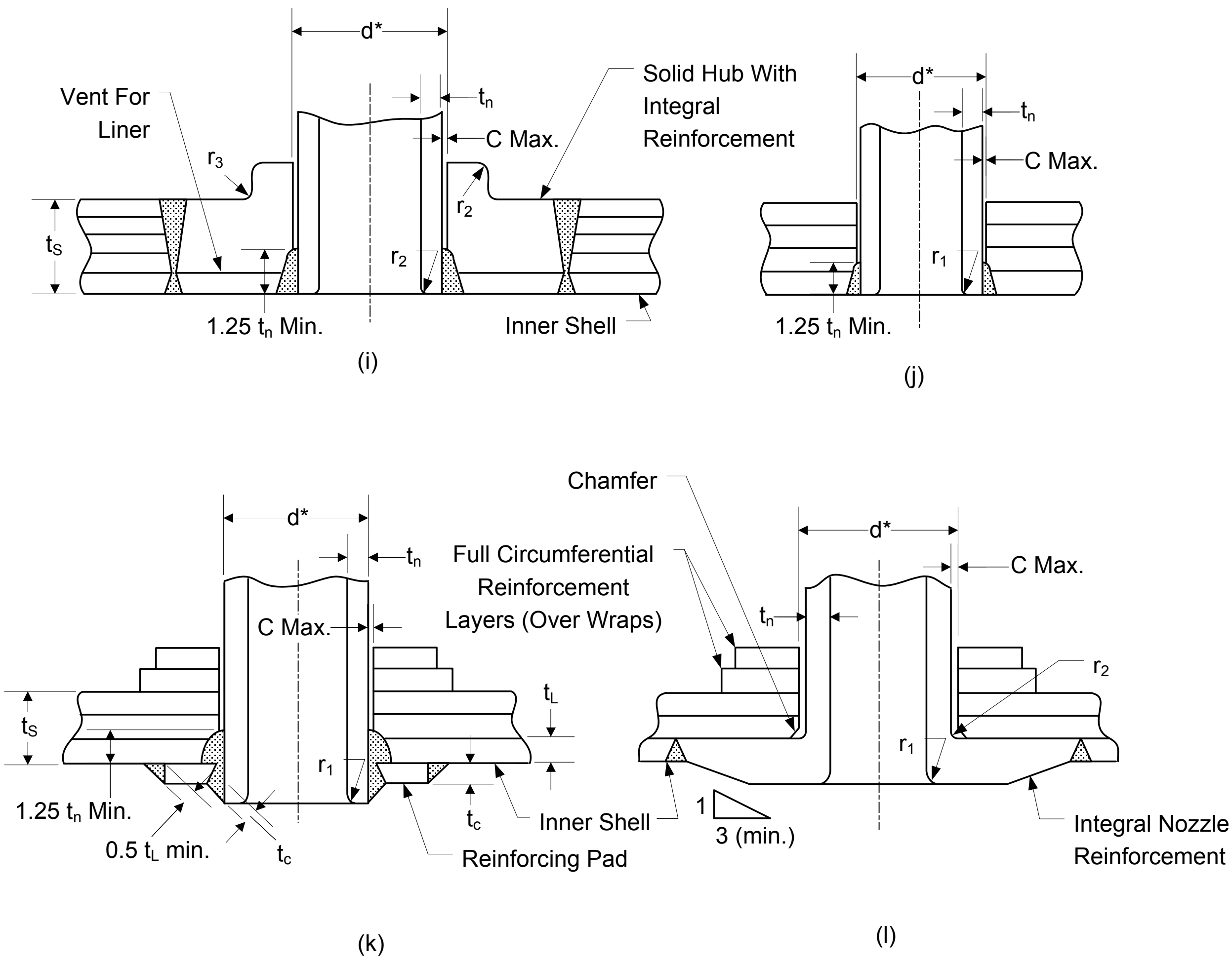


Figure 4.13.9 – Some Acceptable Nozzle Attachments to Layered Shell Sections



Notes:
Provide a means, other than by seal welding, to prevent entry of external foreign matter into the annulus between the layers and the nozzle neck outside diameter for Sketches (i), (j), (k), and (l).

Figure 4.13.9 – Some Acceptable Nozzle Attachments to Layered Shell Sections (Continued)

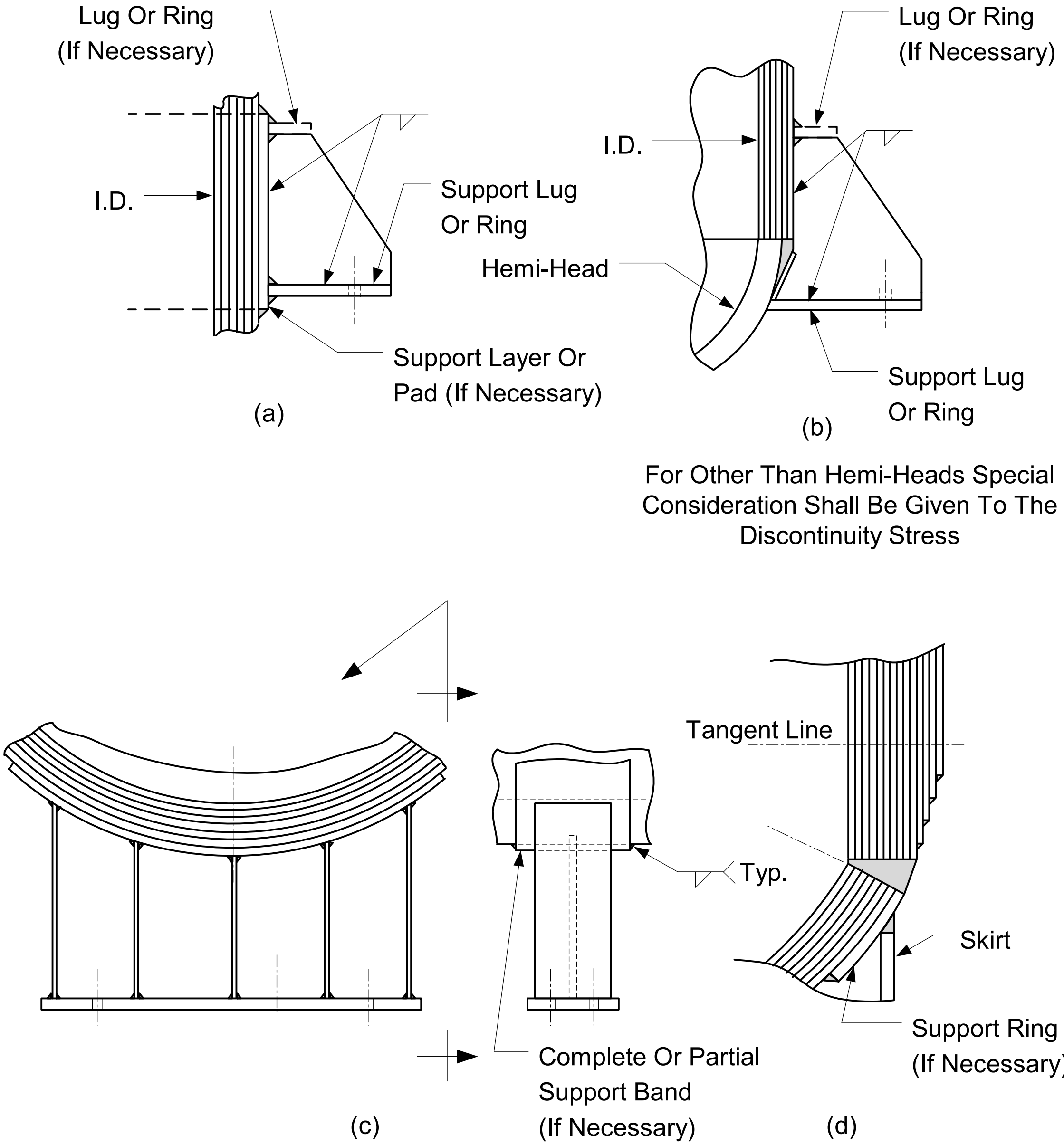


Figure 4.13.10 – Some Acceptable Supports for Layered Vessels

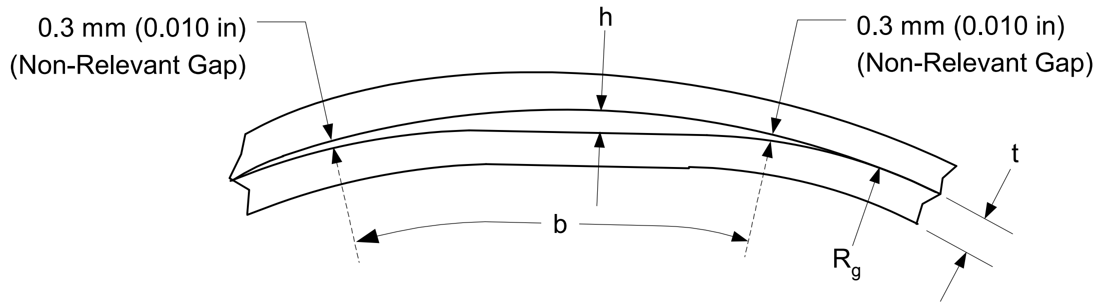


Figure 4.13.11 – Gap Between Vessel Layers

4.14 Evaluation of Vessels Outside of Tolerance

4.14.1 Shell Tolerances

If agreed to by the user, the assessment procedures in Part 5 or in API 579-1/ASME FFS-1 may be used to qualify the design of components that have shell tolerances that do not satisfy the fabrication tolerances in paragraph 4.3.2 and 4.4.4. If API 579-1/ASME FFS-1 is used in the assessment, a Remaining Strength Factor of 0.95 shall be used in the calculations unless another value is agreed to by the user. However, the Remaining Strength Factor shall not be less than 0.90. In addition, a fatigue analysis shall be performed in accordance with API 579-1/ASME FFS-1 as applicable.

4.14.2 Local Thin Areas

4.14.2.1 If agreed to by the user, the assessment procedures in Part 5 or in API 579-1/ASME FFS-1 may be used to qualify the design of components that have a local thin area. A local thin area (LTA) is a region of metal loss on the surface of the component that has a thickness that is less than required by paragraphs 4.3 and 4.4, as applicable. If API 579-1/ASME FFS-1 is used in the assessment, a Remaining Strength Factor of 0.98 shall be used in the calculations unless another value is agreed to by the user. However, the Remaining Strength Factor shall not be less than 0.90. In addition, a fatigue analysis shall be performed in accordance with API 579-1/ASME FFS-1 as applicable.

4.14.2.2 The transition between the LTA and the thicker surface shall be made with a taper length not less than three times the LTA depth. The minimum bottom blend radius shall be equal to or greater than two times the LTA depth (see Figure 4.14.1)

4.14.3 Marking and Reports

The Manufacturer shall maintain records of all calculations including the location and extent of the fabrication tolerances outside the prescribed limits and/or LTAs that are evaluated using paragraph 4.14. This information shall be provided to the user if requested and shall be included in the Manufacturer's Design Report.

4.14.4 Figures

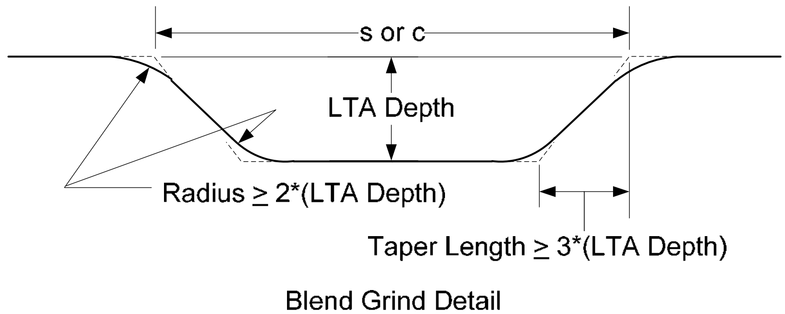


Figure 4.14.1 – LTA Blend Radius Requirements

4.15 Design Rules for Supports and Attachments

4.15.1 Scope

The rules in paragraph 4.15 cover requirements for the design of structural support system(s) for vessels. The structural support system may be, but not limited to, saddles for a horizontal vessel, a skirt for a vertical vessel, or lug and leg type supports for either of these vessel configurations.

4.15.2 Design of Supports

4.15.2.1 Vessels shall be supported for all specified design conditions. The design conditions including load and load case combinations defined in paragraph 4.1.5.3 shall be considered in the design of all vessel supports.

4.15.2.2 Unless otherwise defined in this paragraph, if a stress analysis of the vessel and support attachment configuration is performed, the stress results in the vessel and in the support within the scope of this Division shall satisfy the acceptance criteria in Part 5.

4.15.2.3 The vessel support attachment shall be subject to the fatigue screening criteria of paragraph 5.5.2. In this evaluation, supports welded to the vessel may be considered as integral attachments.

4.15.2.4 All supports shall be designed to prevent excessive localized stresses due to deformations produced by the internal pressure or to thermal gradients in the vessel and support system.

4.15.2.5 Vessel support systems composed of structural steel shapes shall be designed in accordance with a recognized code or standard that cover structural design (e.g. *Specification for Structural Steel Buildings* published by the American Institute of Steel Construction). If the support is at a temperature above ambient due to vessel operation and the recognized code or standard does not provide allowable stresses at temperatures above ambient conditions, then the allowable stress, yield strength, and ultimate tensile strength, as applicable, shall be determined from Annex 3.A and Annex 3.D using a material with a similar minimum specified yield strength and ultimate tensile strength.

4.15.2.6 Attachment welds for structural supports shall be in accordance with paragraph 4.2.

4.15.2.7 Reinforcing plates and saddles attached to the outside of a vessel shall be provided with at least one vent hole that may be tapped for a preliminary compressed air and soap solution (or equivalent) test for tightness of welds that seal the edge of the reinforcing plates and saddles. These vent holes may be left open or may be plugged when the vessel is in service. If the holes are plugged, the plugging material used shall not be capable of sustaining pressure between the reinforcing plate and the vessel wall. Vent holes shall not be plugged during heat treatment.

4.15.2.8 If nonpressure parts such as support lugs, brackets, leg supports and saddles extend over pressure retaining welds, then these welds shall be ground flush for the portion of weld that is covered, or the nonpressure parts shall be notched or coped to clear these welds.

4.15.3 Saddle Supports for Horizontal Vessels

4.15.3.1 Application of Rules

a) Design Method – The design method in this paragraph is based on an analysis of the longitudinal stresses exerted within the cylindrical shell by the overall bending of the vessel, considered as a beam on two single supports, the shear stresses generated by the transmission of the loads on the supports, and the circumferential stresses within the cylindrical shell, the head shear and additional tensile stress in the head, and the possible stiffening rings of this shell, by this transmission of the loads on the supports. The stress calculation method is based on linear elastic mechanics and covers modes of

failure by excessive deformation and elastic instability. Alternatively, saddle supports may be designed in accordance with Part 5.

b) Geometry – A typical horizontal vessel geometry is shown in Figure 4.15.1. Saddle supports for horizontal vessels shall be configured to provide continuous support for at least one-third of the shell circumference, or $\theta \geq 120^0$.

c) Reinforcing Plates – If a reinforcing plate is included in the design to reduce the stresses in the cylindrical shell at the saddle support, then the width of the reinforcing plate, b_1, shall satisfy Equation (4.15.1) and provide a supporting arc length that satisfies Equation (4.15.2). A typical reinforcing plate arrangement is shown in Figure 4.15.2.

$$b_1 = \min\left[\left(b + 1.56\sqrt{R_m t}\right),\ 2a\right] \tag{4.15.1}$$

$$\theta_1 = \theta + \frac{\theta}{12} \tag{4.15.2}$$

d) Stiffening Rings – Stiffening rings may be used at the saddle support location, on either the inside or outside of the cylindrical shell. The stiffening rings may be mounted in the plane of the saddle (see Figure 4.15.3) or two stiffening rings may be mounted on each side of the saddle support equidistant from the saddle support (see Figure 4.15.4). In the later case, the spacing between the two stiffening rings, h, as shown in Figure 4.15.4 shall not be greater than R_m. If $h \leq 1.56\sqrt{R_m t}$ as shown in Figure 4.15.3 Sketch (c), then both of the stiffening rings shall be considered as a single stiffening ring situated in the plane of the saddle in the stress calculations.

4.15.3.2 Moment and Shear Force

a) If the vessel is composed of a cylindrical shell with a formed head (i.e. torispherical, elliptical, or hemispherical) at each end that is supported by two saddle supports equally spaced and with $a \leq 0.25L$, then the moment at the saddle, M_1, the moment at the center of the vessel, M_2, and the shear force at the saddle, T, may be computed using the following equations.

$$M_1 = -Qa\left(1 - \frac{1 - \dfrac{a}{L} + \dfrac{R_m^2 - h_2^2}{2aL}}{1 + \dfrac{4h_2}{3L}}\right) \tag{4.15.3}$$

$$M_2 = \frac{QL}{4}\left(\frac{1 + \dfrac{2\left(R_m^2 - h_2^2\right)}{L^2}}{1 + \dfrac{4h_2}{3L}} - \frac{4a}{L}\right) \tag{4.15.4}$$

$$T = \frac{Q\left(L - 2a\right)}{L + \dfrac{4h_2}{3}} \tag{4.15.5}$$

b) If the vessel supports are not symmetric, or more than two supports are provided, then the highest moment in the vessel, and the moment and shear force at each saddle location shall be evaluated. The

moments and shear force may be determined using strength of materials (i.e. beam analysis with a shear and moment diagram). If the vessel is supported by more than two supports, then differential settlement should be considered in the design.

4.15.3.3 Longitudinal Stress

a) The longitudinal membrane plus bending stresses in the cylindrical shell between the supports are given by the following equations.

$$\sigma_1 = \frac{PR_m}{2t} - \frac{M_2}{\pi R_m^2 t} \qquad (top\ of\ shell) \tag{4.15.6}$$

$$\sigma_2 = \frac{PR_m}{2t} + \frac{M_2}{\pi R_m^2 t} \qquad (bottom\ of\ shell) \tag{4.15.7}$$

b) The longitudinal stresses in the cylindrical shell at the support location are given by the following equations. The values of these stresses depend on the rigidity of the shell at the saddle support. The cylindrical shell may be considered as suitably stiffened if it incorporates stiffening rings at, or on both sides of the saddle support, or if the support is sufficiently close defined as $a \le 0.5R_m$, to a torispherical or elliptical head (a hemispherical head is not considered a stiffening element), a flat cover, or tubesheet.

1) Stiffened Shell – The maximum values of longitudinal membrane plus bending stresses at the saddle support are given by the following equations.

$$\sigma_3 = \frac{PR_m}{2t} - \frac{M_1}{\pi R_m^2 t} \qquad (top\ of\ shell) \tag{4.15.8}$$

$$\sigma_4 = \frac{PR_m}{2t} + \frac{M_1}{\pi R_m^2 t} \qquad (bottom\ of\ shell) \tag{4.15.9}$$

2) Unstiffened Shell – The maximum values of longitudinal membrane plus bending stresses at the saddle support are given by the following equations. The coefficients K_1 and K_1^* are given in Table 4.15.1.

$$\sigma_3^* = \frac{PR_m}{2t} - \frac{M_1}{K_1 \pi R_m^2 t} \qquad (points\ A\ and\ B\ in\ Figure\ 4.15.5) \tag{4.15.10}$$

$$\sigma_4^* = \frac{PR_m}{2t} + \frac{M_1}{K_1^* \pi R_m^2 t} \qquad (bottom\ of\ shell) \tag{4.15.11}$$

c) Acceptance Criteria

1) The absolute value of σ_1, σ_2, and σ_3, σ_4 or σ_3^*, σ_4^*, as applicable shall not exceed SE.

2) If the any of the stresses in paragraph 4.15.3.3.a or 4.15.3.3.b above are negative, the absolute value of the stress shall not exceed S_c that is given by Equation (4.15.12) where $K = 1.0$ for normal operating conditions and $K = 1.35$ for exceptional operating or hydrotest condition.

$$S_c = \frac{KtE_y}{16R_m} \tag{4.15.12}$$

4.15.3.4 Shear Stresses

a) The shear stress in the cylindrical shell with a stiffening ring in the plane of the saddle support is a maximum at Points C and D of Figure 4.15.5 Sketch (b) and shall be computed using Equation (4.15.13).

$$\tau_1 = \frac{T}{\pi R_m t} \tag{4.15.13}$$

b) The shear stress in the cylindrical shell with stiffening rings on both sides of the saddle support is a maximum at Points E and F of Figure 4.15.5 Sketch (c) and shall be computed using Equation (4.15.14). The coefficient K_2 is given in Table 4.15.1.

$$\tau_2 = \frac{K_2 T}{R_m t} \tag{4.15.14}$$

c) The shear stress in a cylindrical shell without stiffening ring(s) that is not stiffened by a formed head, flat cover, or tubesheet, ($a > 0.5R_m$) is also at Points E and F of Figure 4.15.5 Sketch (c) and shall be computed using Equation (4.15.14).

d) The shear stress in the cylindrical shell without stiffening ring(s) and stiffened by a torispherical or elliptical head, flat cover, or tubesheet, ($a \le 0.5R_m$) is a maximum at Points E and F of Figure 4.15.5 Sketch (c) and shall be computed using the equations shown below. In addition to the shear stress, the membrane stress in the formed head, if applicable, shall also be computed using the equations shown below.

1) Shear stress, the coefficient K_3 is given in Table 4.15.1.

$$\tau_3 = \frac{K_3 Q}{R_m t} \qquad (\textit{in the cylindrical shell}) \tag{4.15.15}$$

$$\tau_3^* = \frac{K_3 Q}{R_m t_h} \qquad (\textit{in the formed head}) \tag{4.15.16}$$

2) Membrane stress in a torispherical or elliptical head acting as a stiffener, the coefficient K_4 is given in Table 4.15.1.

$$\sigma_5 = \frac{K_4 Q}{R_m t_h} + \frac{PR_i}{2t_h} \qquad (\textit{torispherical head}) \tag{4.15.17}$$

$$\sigma_5 = \frac{K_4 Q}{R_m t_h} + \frac{PR_i}{2t_h}\left(\frac{R_i}{h_2}\right) \qquad (\textit{elliptical head}) \tag{4.15.18}$$

$$\sigma_5 = 0 \qquad (\textit{flat cover}) \tag{4.15.19}$$

e) Acceptance Criteria

1) The absolute value of τ_1, τ_2, and τ_3, as applicable, shall not exceed $0.8S$ for ferritic materials and $0.6S$ for all other materials.

2) The absolute value of τ_3^* shall not exceed $0.8S_h$ for ferritic materials and $0.6S_h$ for all other materials.

3) The absolute value of σ_5 shall not exceed $1.25S_h$.

4.15.3.5 Circumferential Stress

a) Maximum circumferential bending moment – the distribution of the circumferential bending moment at the saddle support is dependent on the use of stiffeners at the saddle location.

1) Cylindrical shell without a stiffening ring or with a stiffening ring in the plane of the saddle – the maximum circumferential bending moment is shown in Figure 4.15.6 Sketch (a) and shall be computed using Equation (4.15.20). The coefficient K_7 is given in Table 4.15.1.

$$M_\beta = K_7 Q R_m \tag{4.15.20}$$

2) Cylindrical shell with stiffening rings on both side of the saddle – the maximum circumferential bending moment is shown in Figure 4.15.6 Sketch (b) and shall be computed using Equation (4.15.21). The coefficient K_{10} is given in Table 4.15.1.

$$M_\beta = K_{10} Q R_m \tag{4.15.21}$$

b) Width of cylindrical shell – the width of the cylindrical shell that contributes to the strength of the cylindrical shell at the saddle location shall be determined using Equation (4.15.22). If the width x_1 extends beyond the limits in Figures 4.15.2, 4.15.3 or 4.15.4, as applicable, then the width x_1 shall be reduced such as not to exceed this limit.

$$x_1, x_2 \le 0.78\sqrt{R_m t} \tag{4.15.22}$$

c) Circumferential stresses in the cylindrical shell without stiffening ring(s)

1) The maximum compressive circumferential membrane stress in the cylindrical shell at the base of the saddle support shall be computed using Equation (4.15.23). The coefficient K_5 is given in Table 4.15.1.

$$\sigma_6 = \frac{-K_5 Q k}{t\left(b + x_1 + x_2\right)} \tag{4.15.23}$$

2) The circumferential compressive membrane plus bending stress at Points G and H of Figure 4.15.6 Sketch (a) is determined as follows. The coefficient K_7 is given in Table 4.15.1.

   i) If $L \geq 8R_m$, then the circumferential compressive membrane plus bending stress shall be computed using Equation (4.15.24).

$$\sigma_7 = \frac{-Q}{4t\left(b+x_1+x_2\right)} - \frac{3K_7Q}{2t^2} \tag{4.15.24}$$

   ii) If $L < 8R_m$, then the circumferential compressive membrane plus bending stress shall be computed using Equation (4.15.25).

$$\sigma_7^* = \frac{-Q}{4t\left(b+x_1+x_2\right)} - \frac{12K_7QR_m}{Lt^2} \tag{4.15.25}$$

3) The stresses σ_6, σ_7, and σ_7^* may be reduced by adding a reinforcement or wear plate at the saddle location that is welded to the cylindrical shell that satisfies the requirements of paragraph 4.15.3.1.c. The stress can be computed using the equations shown below.

$$\sigma_{6,r} = \frac{-K_5Qk}{b_1\left(t+\eta t_r\right)} \tag{4.15.26}$$

$$\sigma_{7,r} = \frac{-Q}{4\left(t+\eta t_r\right)b_1} - \frac{3K_7Q}{2\left(t+\eta t_r\right)^2} \tag{4.15.27}$$

$$\sigma_{7,r}^* = \frac{-Q}{4\left(t+\eta t_r\right)b_1} - \frac{12K_7QR_m}{L\left(t+\eta t_r\right)^2} \tag{4.15.28}$$

where

$$\eta = \min\left[\frac{S_r}{S},\ 1.0\right] \tag{4.15.29}$$

4) If $t_r > 2t$, then the compressive membrane plus bending stress at the ends of the reinforcing plate (points G_1 and H_1 in Figure 4.15.2 Sketch (b)) shall be computed using the equations shown below. In these equations, coefficient $K_{7,1}$ is computed using the Equation for K_7 in Table 4.15.1 evaluated at the angle θ_1, see Equation (4.15.2).

   i) If $L \geq 8R_m$, then the circumferential compressive membrane plus bending stress shall be computed using Equation (4.15.30)

$$\sigma_{7,1} = \frac{-Q}{4t\left(b+x_1+x_2\right)} - \frac{3K_{7,1}Q}{2t^2} \tag{4.15.30}$$

ii) If $L < 8R_m$, then the circumferential compressive membrane plus bending stress shall be computed using Equation (4.15.31).

$$\sigma^*_{7,1} = \frac{-Q}{4t\left(b+x_1+x_2\right)} - \frac{12K_{7,1}QR_m}{Lt^2} \tag{4.15.31}$$

d) Circumferential stresses in the cylindrical shell with a stiffening ring along the plane of the saddle support.

1) The maximum compressive circumferential membrane stress in the cylindrical shell shall be computed using Equation (4.15.32). The coefficient K_5 is given in Table 4.15.1.

$$\sigma^*_6 = \frac{-K_5Qk}{A} \tag{4.15.32}$$

2) The circumferential compressive membrane plus bending stress at Points G and H of Figure 4.15.6 Sketch (a) for stiffening rings located on the inside of the shell are determined as follows. The coefficients K_8 and K_6 are given in Table 4.15.1.

$$\sigma_8 = \frac{-K_8Q}{A} - \frac{K_6QR_mc_1}{I} \qquad \left(\textit{stress in the shell}\right) \tag{4.15.33}$$

$$\sigma_9 = \frac{-K_8Q}{A} + \frac{K_6QR_mc_2}{I} \qquad \left(\textit{stress in the stiffening ring}\right) \tag{4.15.34}$$

3) The circumferential compressive membrane plus bending stress at Points G and H of Figure 4.15.6 Sketch (a) for stiffening rings located on the outside of the shell are determined as follows. The coefficients K_8 and K_6 are given in Table 4.15.1.

$$\sigma^*_8 = \frac{-K_8Q}{A} + \frac{K_6QR_mc_1}{I} \qquad \left(\textit{stress in the shell}\right) \tag{4.15.35}$$

$$\sigma^*_9 = \frac{-K_8Q}{A} - \frac{K_6QR_mc_2}{I} \qquad \left(\textit{stress in the stiffening ring}\right) \tag{4.15.36}$$

e) Circumferential stresses in the cylindrical shell with stiffening rings on both sides of the saddle support

1) The maximum compressive circumferential membrane stress in the cylindrical shell shall be computed using Equation (4.15.37). The coefficient K_5 is given in Table 4.15.1.

$$\sigma_6 = \frac{-K_5Qk}{t\left(b+2x_2\right)} \tag{4.15.37}$$

2) The circumferential compressive membrane plus bending stress at Points I and J of Figure 4.15.6 Sketch (b) for stiffening rings located on the inside of the shell are determined as follows. The coefficients K_9 and K_{10} are given in Table 4.15.1.

$$\sigma_{10} = \frac{-K_9 Q}{A} + \frac{K_{10} Q R_m c_1}{I} \qquad (\textit{stress in the shell}) \tag{4.15.38}$$

$$\sigma_{11} = \frac{-K_9 Q}{A} - \frac{K_{10} Q R_m c_2}{I} \qquad (\textit{stress in the stiffening ring}) \tag{4.15.39}$$

3) The circumferential compressive membrane plus bending stress at Points I and J of Figure 4.15.6 Sketch (b) for stiffening rings located on the outside of the shell are determined as follows. The coefficients K_9 and K_{10} are given in Table 4.15.1.

$$\sigma_{10}^* = \frac{-K_9 Q}{A} - \frac{K_{10} Q R_m c_1}{I} \qquad (\textit{stress in the shell}) \tag{4.15.40}$$

$$\sigma_{11}^* = \frac{-K_9 Q}{A} + \frac{K_{10} Q R_m c_2}{I} \qquad (\textit{stress in the stiffening ring}) \tag{4.15.41}$$

f) Acceptance Criteria

1) The absolute value of σ_6 or $\sigma_{6,r}$, as applicable, shall not exceed S.

2) The absolute value of σ_6^*, as applicable, shall not exceed $\min[S, S_r]$.

3) The absolute value of σ_7, σ_7^*, $\sigma_{7,r}$, $\sigma_{7,r}^*$, $\sigma_{7,1}$, $\sigma_{7,1}^*$, σ_8, σ_8^*, σ_{10}, and σ_{10}^*, as applicable, shall not exceed $1.25S$.

4) The absolute value of σ_9, σ_9^*, σ_{11}, and σ_{11}^*, as applicable, shall not exceed $1.25S_s$.

4.15.3.6 Saddle Support

The horizontal force at the minimum section at the low point of the saddle is given by Equation (4.15.42). The saddle shall be designed to resist this force.

$$F_h = Q\left(\frac{1+\cos\beta - 0.5\sin^2\beta}{\pi - \beta + \sin\beta\cos\beta}\right) \tag{4.15.42}$$

4.15.4 Skirt Supports for Vertical Vessels

4.15.4.1 The following shall be considered in the design of vertical vessels supported on skirts.

a) The skirt reaction

1) The weight of vessel and contents transmitted in compression to the skirt by the shell above the level of the skirt attachment;

2) The weight of vessel and contents transmitted to the skirt by the weight in the shell below the level of skirt attachment;

3) The load due to externally applied moments and forces when these are a factor, e.g., wind, earthquake, or piping loads.

b) Localized Stresses At The Skirt Attachment Location – High localized stresses may exist in the shell and skirt in the vicinity of the skirt attachment if the skirt reaction is not in line with the vessel wall. When the skirt is attached below the head tangent line, localized stresses are introduced in proportion to the component of the skirt reaction which is normal to the head surface at the point of attachment. When the mean diameter of the skirt and shell approximately coincide (see Figure 4.15.7) and a minimum knuckle radius in accordance with paragraph 4.3 is used, the localized stresses are minimized. In other cases an investigation of local effects may be warranted depending on the magnitude of the loading, location of skirt attachment, etc., and an additional thickness of vessel wall or compression rings may be necessary. Localized stresses at the skirt attachment location may be evaluated by the design by analysis methods in Part 5.

c) Thermal Gradients – Thermal gradients may produce high localized stresses in the vicinity of the vessel to skirt attachment. A "hot-box" detail (see Figure 4.15.8) shall be considered to minimize thermal gradients and localized stresses at the skirt attachment to the vessel wall. If a hot-box is used, the thermal analysis shall consider convection and thermal radiation in the hot-box cavity.

4.15.4.2 The rules of paragraph 4.3.10 shall be used to determine the thickness requirements for the skirt support. Alternatively, skirt supports may be designed using the design by analysis methods in Part 5.

4.15.5 Lug And Leg Supports

4.15.5.1 Lug supports may be used on horizontal or vertical vessels.

4.15.5.2 The localized stresses at the lug support locations on the shell may be evaluated using one of the following methods. If an acceptance criterion is not provided, the results from this analysis shall be evaluated in accordance with Part 5.

a) Part 5 of this Division.

b) Welding Research Council Bulletin Number 107, *Local Stresses in Spherical and Cylindrical Shells Due to External Loadings*.

c) Welding Research Council Bulletin 198, Part 1, *Secondary Stress Indices for Integral Structural Attachments to Straight Pipes*; Part 2, *Stress Indices at Lug Supports on Piping Systems*.

d) Welding Research Council Bulletin 353, *Position Paper On Nuclear Plant Pipe Supports.*

e) Welding Research Council Bulletin 448, *Evaluation of Welded Attachments on Pipe and Elbows*.

f) Other analytical methods contained in recognized codes and standards for pressure vessel construction (i.e. British Standard PD-5500, *Specification for Fusion Welded Pressure Vessels (Advanced Design and Construction) for Use in the Chemical, Petroleum, and Allied Industries).*

4.15.5.3 If vessels are supported by lugs, legs, or brackets attached to the shell, then the supporting members under these bearing attachments should be as close to the shell as possible to minimize local bending stresses in the shell.

4.15.5.4 Supports, lugs, brackets, stiffeners, and other attachments may be attached with stud bolts to the outside or inside of a vessel wall.

4.15.5.5 Lug and column supports should be located away from structural discontinuities (i.e. cone-to-cylinder junctions) and Category A or B weld seams. If these supports are located within $1.8\sqrt{Dt}$ of these locations, then a stress analysis shall be performed and the results from this analysis shall be evaluated in accordance with paragraph 4.15.5.2.

4.15.6 Nomenclature

A	cross-sectional area of the stiffening ring(s) and the associated shell width used in the stress calculation.
a	distance from the axis of the saddle support to the tangent line on the curve for a dished head or to the inner face of a flat cover or tubesheet.
b	width of contact surface of the cylindrical shell and saddle support.
b_1	width of the reinforcing plate welded to the cylindrical shell at the saddle location
c_1, c_2	distance to the extreme axes of the cylinder-stiffener cross section to the neutral axis of the cylinder-stiffener cross-section
E_y	modulus of elasticity.
E	weld joint efficiency (see paragraph 4.2.4) for the circumferential weld seam being evaluated.
η	shell to reinforcing plate strength reduction factor.
F_h	saddle horizontal force.
h	spacing between two mounted stiffening rings placed o each side of the saddle support.
h_2	depth of the elliptical head.
I	moment of inertia of cross-sectional area A in relation to its neutral axis that is parallel to the axis of the cylindrical shell.
k	factor to account for the vessel support condition; $k = 1$ is the vessel is resting on the support and $k = 0.1$ is the vessel is welded to the support.
K	factor to set the allowable compressive stress for the cylindrical shell material.
L	length of the cylindrical shell measured from tangent line to tangent line for a vessel with dished heads or from the inner face to inner face for vessels with flat covers or tubesheets.
M_1	net-section maximum longitudinal bending moment at the saddle support; this moment is negative when it results in a tensile stress on the top of the shell.
M_2	net-section maximum longitudinal bending moment between the saddle supports; this moment is positive when it results in a compressive stress on the top of the shell.
P	design pressure, positive for internal pressure and negative for external pressure.
Q	maximum value of the reaction at the saddle support from weight and other loads as applicable.
R_i	inside radius of the spherical dome or a torispherical head.
R_m	mean radius of the cylindrical shell.
S	allowable stress from Annex 3.A for the cylindrical shell material at the design temperature.
S_c	allowable compressive stress for the cylindrical shell material at the design temperature.
S_h	allowable stress from Annex 3.A for the head material at the design temperature.
S_r	allowable stress from Annex 3.A for the reinforcing plate material at the design temperature.
S_s	allowable stress from Annex 3.A for the stiffener material at the design temperature.
t	cylindrical shell or shell thickness, as applicable.
t_h	head thickness.
t_r	reinforcing plate thickness.
T	maximum shear force at the saddle.
θ	opening of the supported cylindrical shell arc.
θ_1	opening of the cylindrical shell arc engaged by a welded reinforcing plate.

x_1, x_2 width of cylindrical shell used in the circumferential normal stress strength calculation.

4.15.7 Tables

Table 4.15.1 – Stress Coefficients For Horizontal Vessels on Saddle Supports

Stress Coefficient
$K_1 = \dfrac{\Delta + \sin\Delta \cdot \cos\Delta - \dfrac{2\sin^2\Delta}{\Delta}}{\pi\left(\dfrac{\sin\Delta}{\Delta} - \cos\Delta\right)}$
$K_1^* = \dfrac{\Delta + \sin\Delta \cdot \cos\Delta - \dfrac{2\sin^2\Delta}{\Delta}}{\pi\left(1 - \dfrac{\sin\Delta}{\Delta}\right)}$
$K_2 = \dfrac{\sin\alpha}{\pi - \alpha + \sin\alpha \cdot \cos\alpha}$
$K_3 = \left(\dfrac{\sin\alpha}{\pi}\right)\left(\dfrac{\alpha - \sin\alpha \cdot \cos\alpha}{\pi - \alpha + \sin\alpha \cdot \cos\alpha}\right)$
$K_4 = \dfrac{3}{8}\left(\dfrac{\sin^2\alpha}{\pi - \alpha + \sin\alpha \cdot \cos\alpha}\right)$
$K_5 = \dfrac{1 + \cos\alpha}{\pi - \alpha + \sin\alpha \cdot \cos\alpha}$
$K_6 = \dfrac{\dfrac{3\cos\beta}{4}\left(\dfrac{\sin\beta}{\beta}\right)^2 - \dfrac{5\sin\beta\cos^2\beta}{4\beta} + \dfrac{\cos^3\beta}{2} - \dfrac{\sin\beta}{4\beta} + \dfrac{\cos\beta}{4} - \beta\sin\beta\left[\left(\dfrac{\sin\beta}{\beta}\right)^2 - \dfrac{1}{2} - \dfrac{\sin 2\beta}{4\beta}\right]}{2\pi\left[\left(\dfrac{\sin\beta}{\beta}\right)^2 - \dfrac{1}{2} - \dfrac{\sin 2\beta}{4\beta}\right]}$
$K_7 = \dfrac{K_6}{4} \qquad when\ \dfrac{a}{R_m} \le 0.5$
$K_7 = \dfrac{3}{2}K_6\left(\dfrac{a}{R_m}\right) - \dfrac{1}{2}K_6 \qquad when\ 0.5 < \dfrac{a}{R_m} < 1$
$K_7 = K_6 \qquad when\ \dfrac{a}{R_m} \ge 1$

Table 4.15.1 – Stress Coefficients For Horizontal Vessels on Saddle Supports

Stress Coefficient
$K_8 = \dfrac{\cos\beta\left[1 - \dfrac{\cos 2\beta}{4} + \dfrac{9\sin\beta\cos\beta}{4\beta} - 3\left(\dfrac{\sin\beta}{\beta}\right)^2\right]}{2\pi\left[\left(\dfrac{\sin\beta}{\beta}\right)^2 - \dfrac{1}{2} - \dfrac{\sin 2\beta}{4\beta}\right]} + \dfrac{\beta\sin\beta}{2\pi}$ $K_9 = \dfrac{1}{2\pi}\left\{\left[-\dfrac{1}{2} + (\pi - \beta)\cot\beta\right]\cos\rho + \rho\sin\rho\right\}$ $K_{10} = \dfrac{1}{2\pi}\left\{\rho\sin\rho + \cos\rho\left[\dfrac{3}{2} + (\pi - \beta)\cot\beta\right] - \dfrac{(\pi - \beta)}{\sin\beta}\right\}$

Notes:

1. $\Delta = \dfrac{\pi}{6} + \dfrac{5\theta}{12}$
2. $\alpha = 0.95\left(\pi - \dfrac{\theta}{2}\right)$
3. $\beta = \pi - \dfrac{\theta}{2}$
4. The relationship between ρ and θ is given by $\rho = \tan\rho\left[0.5 + (\pi - \beta)\cot\beta\right]$. Values for ρ for a specified θ are shown in the table below.

Relationship Between ρ and θ							
θ	120°	130°	140°	150°	160°	170°	180°
ρ	93.667°	91.133°	87.833°	84.167°	79.667°	74°	66.933°

Note: $\rho = -158.58 + 7.8668\theta - 8.8037(10)^{-2}\theta^2 + 4.3011(10)^{-4}\theta^3 - 8.0644(10)^{-7}\theta^4$ for all values of θ that satisfy $120^\circ \le \theta \le 180^\circ$. This curve fit provides ρ in degrees.

5. The angles Δ, θ, β, and ρ are in radians in the calculations.

4.15.8 Figures

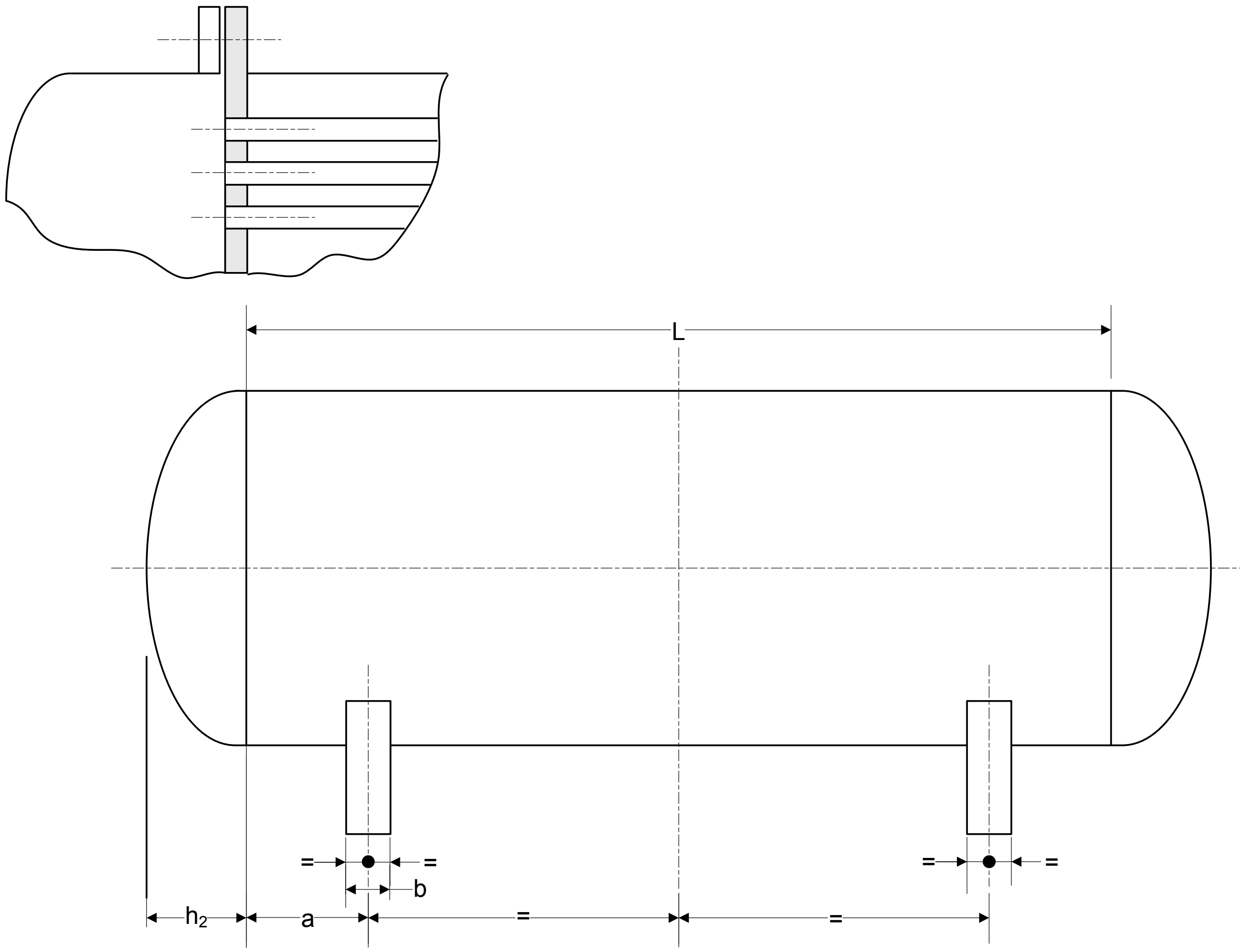


Figure 4.15.1 – Horizontal Vessel on Saddle Supports

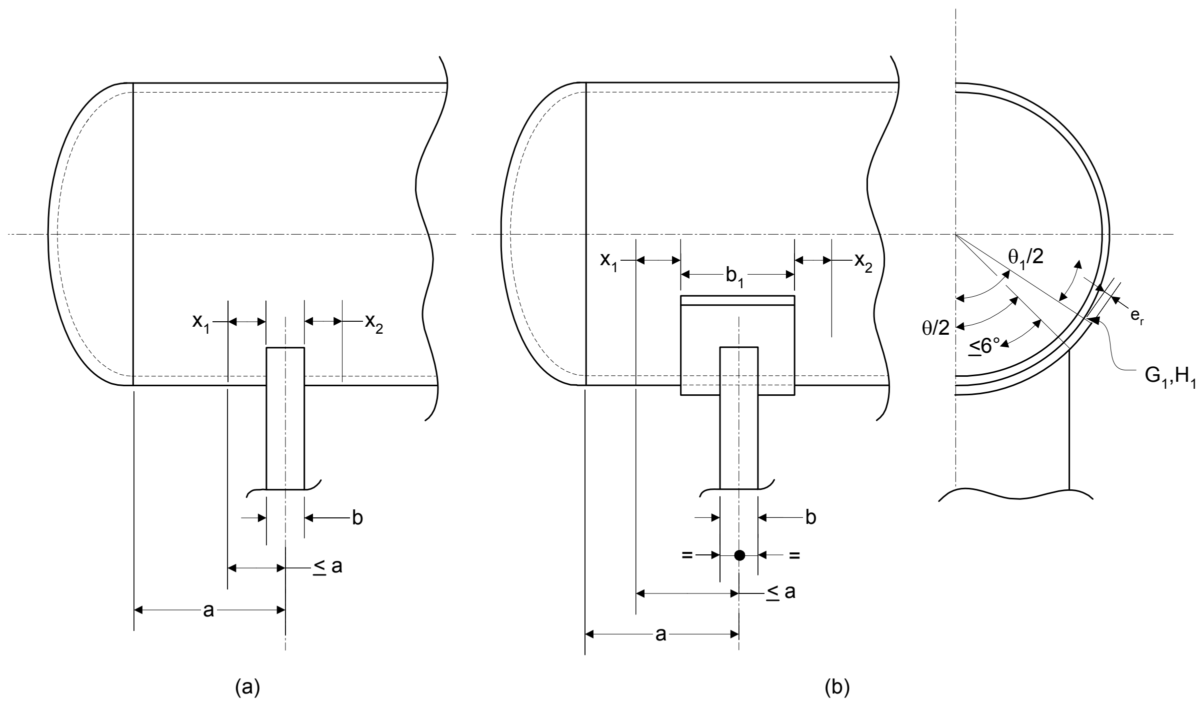


Figure 4.15.2 – Cylindrical Shell Without Stiffening Rings

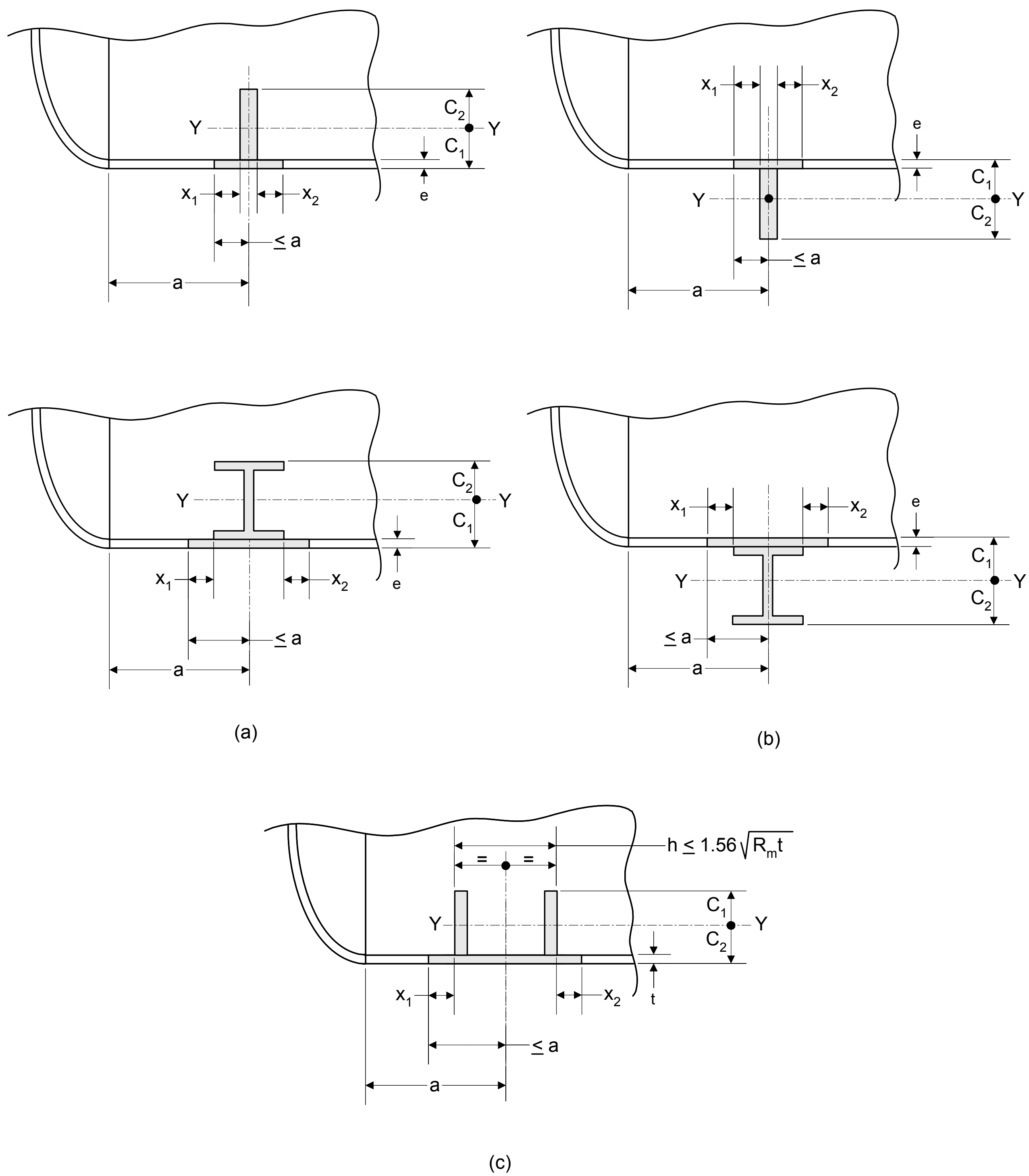


Figure 4.15.3 – Cylindrical Shell With Stiffening Rings in the Plane of the Saddle

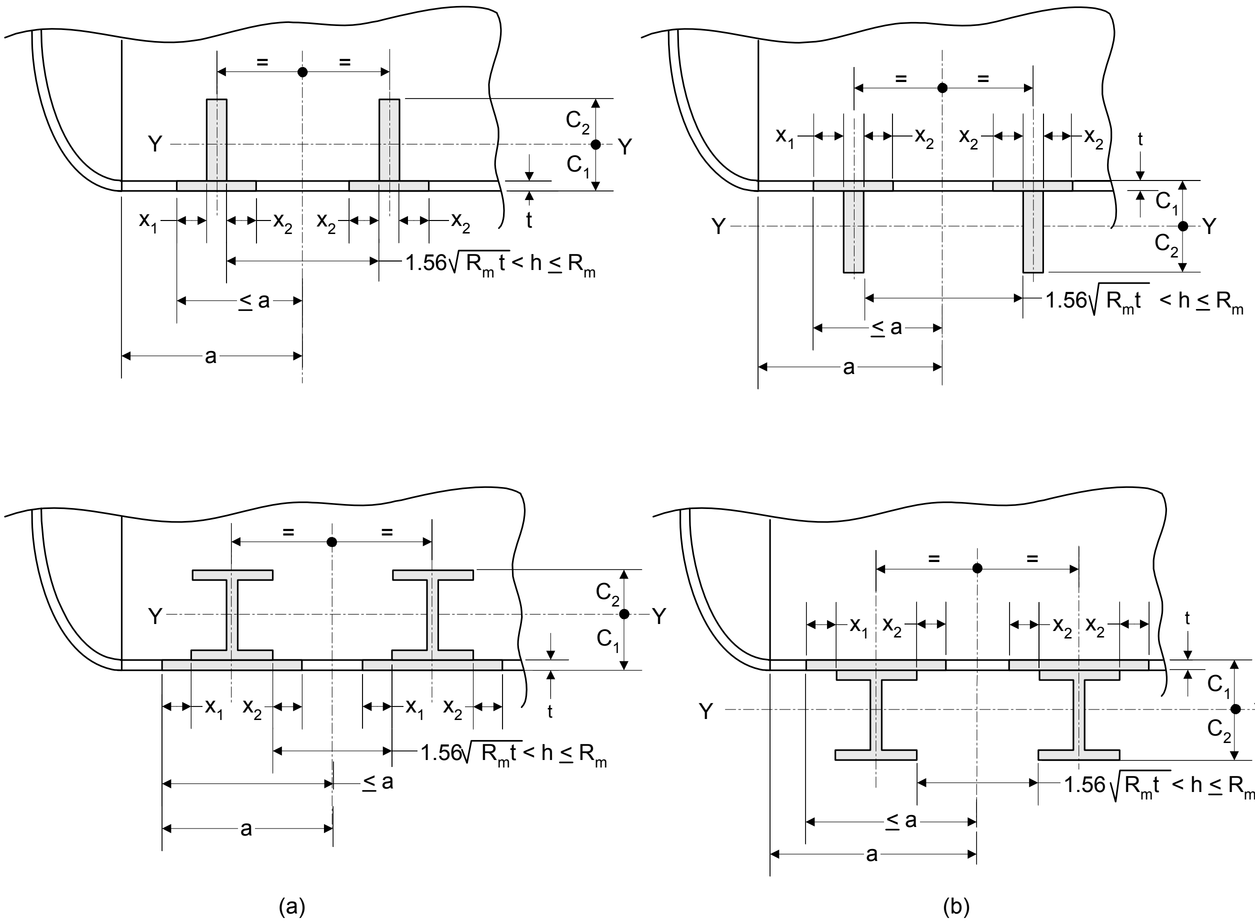


Figure 4.15.4 – Cylindrical Shell With Stiffening Rings on Both Sides of the Saddle

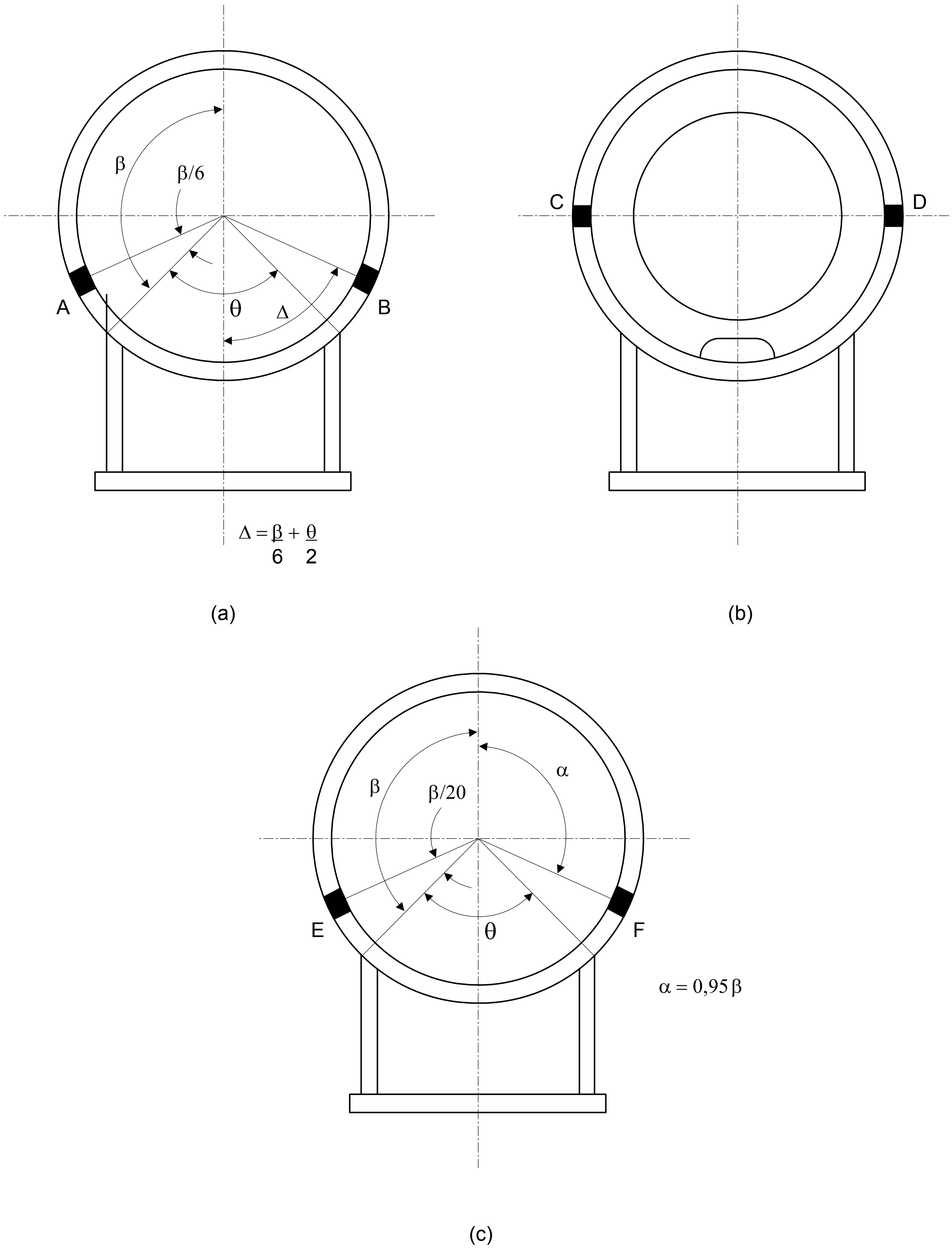
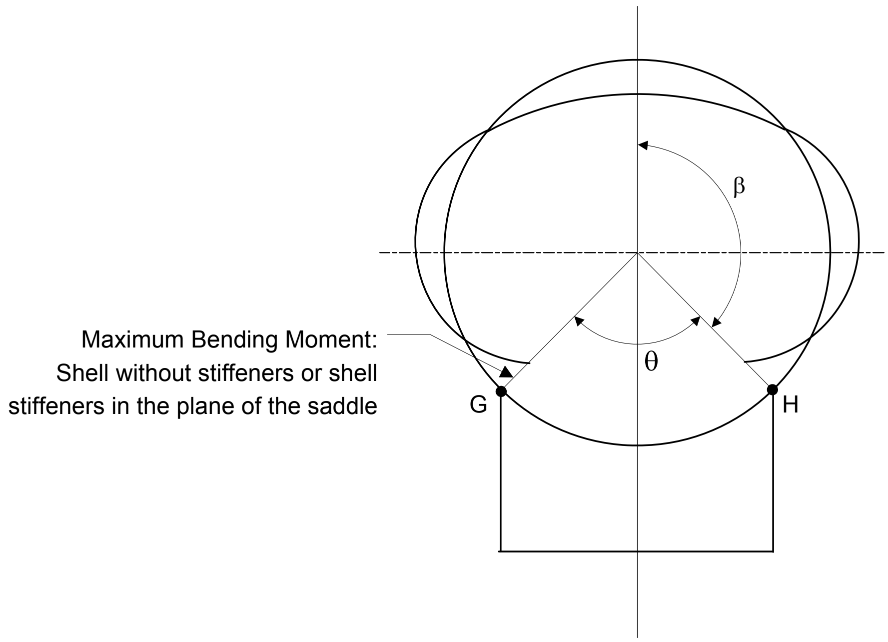
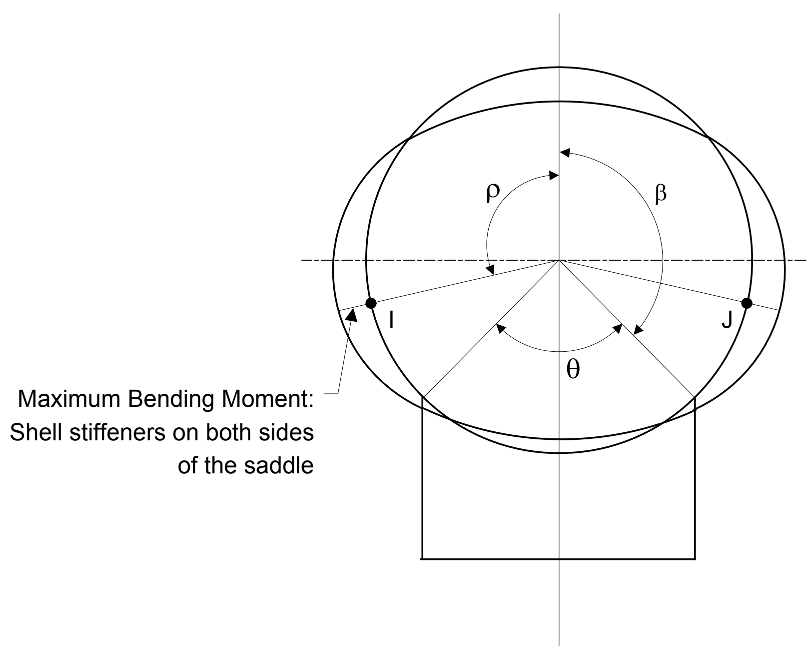


Figure 4.15.5 – Locations of Maximum Longitudinal Normal Stress and Shear Stress in the Cylinder



(a)



(b)

Figure 4.15.6 – Locations of Maximum Circumferential Normal Stresses in the Cylinder

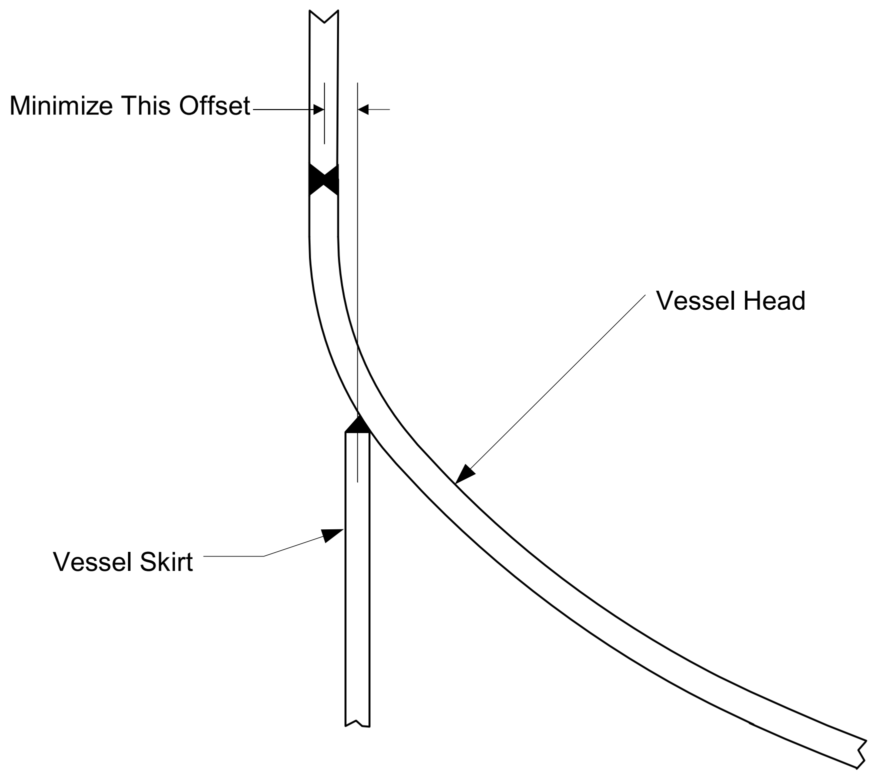


Figure 4.15.7 – Skirt Attachment Location on Vertical Vessels

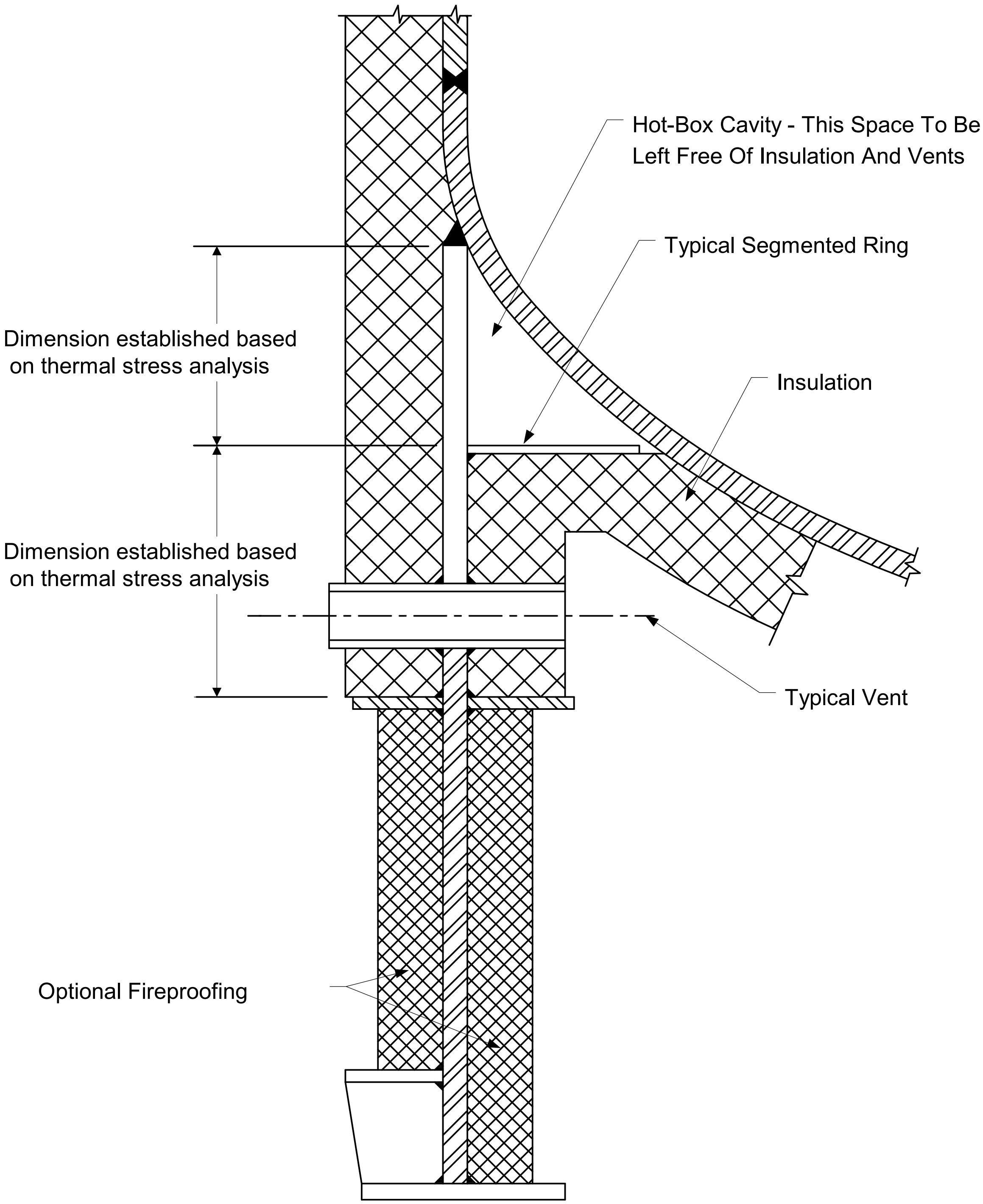


Figure 4.15.8 – A Typical Hot-Box Arrangement for Skirt Supported Vertical Vessels

4.16 Design Rules for Flanged Joints

4.16.1 Scope

4.16.1.1 The rules in paragraph 4.16 shall be used to design circular flanges subject to internal and/or external pressure. These rules provide for hydrostatic end loads, gasket seating, and externally applied axial force and net-section bending moment.

4.16.1.2 The rules in paragraph 4.16 apply to the design of bolted flange connections with gaskets that are entirely located within the circle enclosed by the bolt holes. The rules do not cover the case where the gasket extends beyond the bolt hole circle or where metal-metal contact is made outside of the bolt circle.

4.16.1.3 It is recommended that bolted flange connections conforming to the standards listed in paragraph 4.1.11 be used for connections to external piping. These standards may be used for other bolted flange connections and dished covers within the limits of size in the standard and pressure-temperature ratings permitted in paragraph 4.1.11. The ratings in these standards are based on the hub dimensions given or on the minimum specified thickness of flanged fittings of integral construction. Flanges fabricated from rings may be used in place of the hub flanges in these standards provided that their strength and rigidity, calculated by the rules in this paragraph, are not less than that calculated for the corresponding size of hub flange.

4.16.1.4 The rules of this paragraph should not be construed to prohibit the use of other types of flanged connections provided they are designed in accordance with Part 5.

4.16.2 Design Considerations

4.16.2.1 The design of a flange involves the selection of the flange type, gasket material, flange facing, bolting, hub proportions, flange width, and flange thickness. The flange dimensions shall be selected such that the stresses in the flange and the flange rigidity satisfy the acceptability criteria of this paragraph.

4.16.2.2 In the design of a bolted flange connection, calculations shall be made for the following two design conditions, and the most severe condition shall govern the design of the flanged joint.

a) *Operating Conditions* – The conditions required to resist the hydrostatic end force of the design pressure and any applied external forces and moments tending to part the joint at the design temperature.

b) *Gasket Seating Condition* – The conditions existing when the gasket or joint-contact surface is seated by applying an initial load with the bolts during assembly of the joint, at atmospheric temperature and pressure.

4.16.2.3 Calculations shall be performed using dimensions of the flange in the corroded and uncorroded conditions.

4.16.2.4 In the design of flange pairs, each flange is designed for its particular design loads of pressure and gasket reactions. The bolt load used to design each flange, however, is that load common to the flange pair and equal to the larger of the bolt loads calculated for each flange individually. No additional rules are required for design of flange pairs. After the loads for the most severe condition are determined, calculations shall be made for each flange following the rules of this paragraph.

4.16.2.5 In the design of flange pairs where pass partitions with gaskets are used, the gasket loads from the partition(s) shall be included in the calculation of bolt loads. Partition gaskets may have different gasket constants than the ring gasket inside the bolt circle. In the design of flanges with noncircular gaskets or with partitions of any shape, gasket reactions from all surfaces with gaskets shall be included in calculating bolt loads.

4.16.3 Flange Types

4.16.3.1 For the purpose of computation, there are two major categories of flanges:

a) Integral Type Flanges – This type covers designs where the flange is cast or forged integrally with the nozzle neck, vessel or pipe wall, butt welded thereto, or attached by other forms of welding such that the flange and nozzle neck, vessel or pipe wall are structurally equivalent to integral construction. Integral flanges shall be designed considering structural interaction between the flange and the nozzle neck, vessel, or pipe wall, which the rules account for by considering the neck or wall to act as a hub. Integral type flanges are referenced below. The design flange and bolt loads are shown in Figures 4.16.1 and 4.16.2.

   1) Integral type flanges – Figure 4.16.1 Sketch (a) and Table 4.2.9, Details 9 and 10

   2) Integral type flanges where $g_1 = g_o$ – Figure 4.16.1 Sketch (b)

   3) Integral type flanges with a hub – Figure 4.16.2 and Table 4.2.9, Details 6, 7, and 8

   4) Integral type flanges with nut stops – Figure 4.16.3 and Figure 4.16.4

b) Loose Type Flanges – This type covers those designs in which the flange has no substantial integral connection to the nozzle neck, vessel, or pipe wall, and includes welded flange connections where the welds are not considered to give the mechanical strength equivalent of an integral attachment. Loose type flanges are referenced below. The design flange and bolt loads are shown in Figures 4.16.5 and 4.16.6.

   1) Loose type flanges – Figure 4.16.5 and Table 4.2.9, Details 1,2,3 and 4

   2) Loose type lap joint flanges – Figure 4.16.6 and Table 4.2.9, Detail 5

4.16.3.2 The integral and loose type flanges described above can also be applied to reverse flange configurations. Integral and loose type reverse flanges are shown in Figure 4.16.7.

4.16.4 Flange Materials

4.16.4.1 Materials used in the construction of bolted flange connections, excluding gasket materials, shall comply with the requirements given in Part 3.

4.16.4.2 Flanges made from ferritic steel shall be given a normalizing or full-annealing heat treatment when the thickness of the flange section exceeds 75 mm (3 in.).

4.16.4.3 Fabricated flanges with hub shall be in accordance with the following:

a) Flanges with hubs may be machined from a hot rolled or forged billet or forged bar. The axis of the finished flange shall be parallel to the long axis of the original billet or bar, but these axes need not be concentric.

b) Flanges with hubs, except as permitted in paragraph 4.16.4.3.a, shall not be machined from plate or bar stock material unless the material has been formed into a ring, and further provided that:

   1) In a ring formed from plate, the original plate surfaces are parallel to the axis of the finished flange;

   2) The joints in the ring are welded butt joints that conform to the requirements of Part 6. The thickness to be used to determine postweld heat treatment and radiographic requirements shall be $\min\left[t, \left(A-B\right)/2\right]$.

c) The back of the flange and the outer surface of the hub shall be examined by either the magnetic particle method or the liquid penetrant method in accordance with Part 7.

4.16.4.4 Bolts, studs, nuts, and washers shall comply with the requirements of Part 3 and referenced standards. It is recommended that bolts and studs have a nominal diameter of not less than 12 mm (0.5 in.). If bolts or studs smaller than 12 mm (0.5 in.) are used, then ferrous bolting material shall be of alloy steel. Precautions shall be taken to avoid overstressing small-diameter bolts. When washers are used, they shall be through hardened to minimize the potential for galling.

4.16.5 Gasket Materials

4.16.5.1 The gasket constants for the design of the bolt load (m and y), are provided in Table 4.16.1. Other values for the gasket constants may be used if based on actual testing or data in the literature, as agreed upon between designer and the user.

4.16.5.2 The minimum width of sheet and composite gaskets, N , is recommended to be no less than that given in Table 4.16.2.

NOTE: Gasket materials should be selected that are suitable for the design conditions. Corrosion, chemical attack, creep and thermal degradation of gasket materials over time should be considered.

4.16.6 Design Bolt Loads

4.16.6.1 The procedure to determine the bolt loads for the operating and gasket seating conditions is shown below.

a) STEP 1 – Determine the design pressure and temperature of the flange joint.

b) STEP 2 – Select a gasket and determine the gasket factors m and y from Table 4.16.1, or other sources. The selected gasket width should comply with the guidelines detailed in Table 4.16.2.

c) STEP 3 – Determine the width of the gasket, N , basic gasket seating width, b_0, the effective gasket seating width, b , and the location of the gasket reaction, G , based on the flange and gasket geometry, the information in Table 4.16.3 and Figure 4.16.8, and the equations shown below. Note that for lap joint flanges, G is equal to the midpoint of contact between the flange and the lap, see Figure 4.16.6 and Figure 4.16.8.

1) For $b_0 \le 6\,mm\ (0.25\ in.)$, G is the mean diameter of the gasket contact face and

$$b = b_0 \tag{4.16.1}$$

2) For $b_0 > 6\,mm\ (0.25\ in.)$

$$b = 0.5C_{ul}\sqrt{\frac{b_0}{C_{ul}}} \tag{4.16.2}$$

$$G = G_C - 2b \tag{4.16.3}$$

d) STEP 4 – Determine the design bolt load for the operating condition.

$$W_o = 0.785G^2P + 2b\pi GmP \qquad for\ non-self-energized\ gaskets \tag{4.16.4}$$

$$W_o = 0.785G^2P \qquad for\ self-energized\ gaskets \tag{4.16.5}$$

e) STEP 5 – Determine the design bolt load for the gasket seating condition.

$$W_g = \left(\frac{A_m + A_b}{2}\right) S_{bg} \tag{4.16.6}$$

The parameter A_b is the actual total cross sectional area of the bolts that is selected such that $A_b \ge A_m$, where:

$$A_m = \max\left[\left(\frac{W_o + F_A + \dfrac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_{gs}}{S_{bg}}\right)\right] \tag{4.16.7}$$

$$W_{gs} = \pi b G\left(C_{us} y\right) \qquad \textit{for non-self-energized gaskets} \tag{4.16.8}$$

$$W_{gs} = 0.0 \qquad \textit{for self-energized gaskets} \tag{4.16.9}$$

Note: Where significant axial force is required to compress the gasket during assembly of a joint containing a self-energizing gasket, the value of W_{gs} shall be taken as equal to that axial force. In addition, some self-energizing gaskets generate axial load due to their wedging action and this load shall be considered in setting the value of W_{gs}.

4.16.7 Flange Design Procedure

4.16.7.1 The procedure in this paragraph can be used to design circular integral, loose or reverse flanges, subject to internal or external pressure, and external loadings. The procedure incorporates both a strength check and a rigidity check for flange rotation.

4.16.7.2 The procedure to design a flange is shown below.

a) This STEP 1 – Determine the design pressure and temperature of the flange joint, and the external net-section axial force, F_A, and bending moment, M_E. If the pressure is negative, the absolute value of the pressure should be used in this procedure.

b) STEP 2 – Determine the design bolt loads for operating condition, W_o, and the gasket seating condition, W_g, and corresponding actual bolt area, A_b, from paragraph 4.16.6.

c) STEP 3 – Determine an initial flange geometry, in addition to the information required to determine the bolt load, the following geometric parameters are required:

1) The flange bore, B
2) The bolt circle diameter, C
3) The outside diameter of the flange, A
4) The flange thickness, t
5) The thickness of the hub at the large end, g_1
6) The thickness of the hub at the small end, g_0

7) The hub length, h

d) STEP 4 – Determine the flange stress factors using the equations in Tables 4.16.4 and 4.16.5.

e) STEP 5 – Determine the flange forces.

$$H_D = 0.785B^2P \tag{4.16.10}$$

$$H = 0.785G^2P \tag{4.16.11}$$

$$H_T = H - H_D \tag{4.16.12}$$

$$H_G = W_o - H \tag{4.16.13}$$

f) STEP 6 – Determine the flange moment for the operating condition using Equation (4.16.14) or Equation (4.16.15), as applicable. When specified by the user or his designated agent, the maximum bolt spacing ($B_{s\max}$) and the bolt spacing correction factor (B_{sc}) shall be applied in calculating the flange moment for internal pressure using the equations in Table 4.16.11. The flange moment M_o for the operating condition and flange moment M_g for the gasket seating condition without correction for bolt spacing $B_{sc} = 1$ is used for the calculation of the rigidity index in STEP 10. In these equations, h_D, h_T, and h_G are determined from Table 4.16.6. For integral and loose type flanges, the moment M_{oe} is calculated using Equation (4.16.16) where I and I_p in this equation are determined from Table 4.16.7. For reverse type flanges, the procedure to determine M_{oe} shall be agreed upon between the Designer and the Owner.

$$M_o = \text{abs}\left[\left((H_D h_D + H_T h_T + H_G h_G)B_{sc} + M_{oe}\right)F_s\right] \quad \textit{for internal pressure} \tag{4.16.14}$$

$$M_o = \text{abs}\left[\left(H_D\left(h_D - h_G\right) + H_T\left(h_T - h_G\right) + M_{oe}\right)F_s\right] \quad \textit{for external pressure} \tag{4.16.15}$$

$$M_{oe} = 4M_E\left[\frac{I}{0.3846I_p + I}\right]\left[\frac{h_D}{\left(C - 2h_D\right)}\right] + F_A h_D \tag{4.16.16}$$

g) STEP 7 – Determine the flange moment for the gasket seating condition using Equation (4.16.17) or Equation (4.16.18), as applicable.

$$M_g = \frac{W_g\left(C - G\right)B_{sc}F_s}{2} \quad \textit{for internal pressure} \tag{4.16.17}$$

$$M_g = W_g h_G F_s \quad \textit{for external pressure} \tag{4.16.18}$$

h) STEP 8 – Determine the flange stresses for the operating and gasket seating conditions using the equations in Table 4.16.8.

i) STEP 9 – Check the flange stress acceptance criteria. The two criteria shown below shall be evaluated. If the stress criteria are satisfied, go to STEP 10. If the stress criteria are not satisfied, re-proportion the flange dimensions and go to STEP 4.

1) Allowable Normal Stress – The criteria to evaluate the normal stresses for the operating and gasket seating conditions are shown in Table 4.16.9.

2) Allowable Shear Stresses – In the case of loose type flanges with lap, as shown in Fig. 4.16.6 where the gasket is so located that the lap is subjected to shear, the shearing stress shall not exceed $0.8S_{no}$ or $0.8S_{ng}$, as applicable, for the material of the lap. In the case of welded flanges where the nozzle neck, vessel, or pipe wall extends near to the flange face and may form the gasket contact face, the shearing stress carried by the welds shall not exceed $0.8S_{no}$ or $0.8S_{ng}$, as applicable. The shearing stress shall be calculated for both the operating and gasket seating load cases. Similar situations where flange parts are subjected to shearing stresses shall be checked using the same requirement.

j) STEP 10 – Check the flange rigidity criterion in Table 4.16.10. If the flange rigidity criterion is satisfied, then the design is complete. If the flange rigidity criterion is not satisfied, then re-proportion the flange dimensions and go to STEP 3. The flange moment M_o for the operating condition (STEP 6) and flange moment M_g for the gasket seating condition (STEP 7) without correction for bolt spacing $B_{sc}=1$ is used for the calculation of the rigidity index.

4.16.8 Split Loose Type Flanges

Loose flanges split across a diameter and designed under the rules given in this paragraph may be used under the following provisions.

a) When the flange consists of a single split flange or flange ring, it shall be designed as if it were a solid flange (without splits), using 200% of the total moment, $F_s = 2.0$.

b) When the flange consists of two split rings, each ring shall be designed as if it were a solid flange (without splits), using 75% of the total moment, $F_s = 0.75$. The pair of rings shall be assembled so that the splits in one ring are 90 degrees from the splits in the other ring.

c) The flange split locations should preferably be midway between bolt holes.

4.16.9 Noncircular Shaped Flanges with a Circular Bore

The outside diameter, A, for a noncircular flange with a circular bore shall be taken as the diameter of the largest circle, concentric with the bore, inscribed entirely within the outside edges of the flange. The bolt loads, flange moments, and stresses shall be calculated in the same manner as for a circular flange using a bolt circle whose size is established by drawing a circle through the centers of the outermost bolts.

4.16.10 Flanges with Nut Stops

When flanges are designed per paragraph 4.16, or are fabricated to the dimensions of ASME B16.5 or other acceptable standards, except that the dimension $0.5(C-B)-g_1$ is decreased to provide a nut-stop, the fillet radius shall be as shown in Figures 4.16.3 and 4.16.4 except that:

a) For flanges designed to this paragraph, the thickness of the hub at the large end, g_1, shall be the smaller of $2t_n$ or $4r_u$, but not less than 12 mm (0.5 in.).

b) For ASME B16.5 or other standard flanges, the thickness of the hub at the small end, g_0, shall be increased as necessary to provide a nut-stop.

4.16.11 Joint Assembly Procedures

Bolted joints should be assembled and bolted-up in accordance with a written procedure that has been demonstrated to be acceptable for similar joint configurations in similar services. Further guidance can be found in ASME PCC-1 "Guidelines for Pressure Boundary Bolted Flange Joint Assembly".

4.16.12 Nomenclature

A	outside diameter of the flange or, where slotted holes extend to the outside of the flange, the diameter to the bottom of the slots.
A_b	cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion.
A_m	total minimum required cross-sectional area of the bolts.
a	nominal bolt diameter
B	inside diameter of the flange. When $B < 20g_1$, B_1 may be used for B in the equation for the longitudinal stress.
B_1	$B + g_1$ for loose type flanges and for integral type flanges that have a value of f less than 1.0, (although a minimum value of $f = 1.0$ is permitted). B_1 is equal to $B + g_0$ for integral type flanges when $f \geq 1.0$.
B^*	inside diameter of the reverse flange.
B_s	bolt spacing, The bolt spacing may be taken as the bolt circle circumference divided by the number of bolts or as the chord length between adjacent bolt locations.
$B_{s\max}$	maximum bolt spacing
B_{sc}	bolt spacing correction factor
b	effective gasket contact width.
b_0	basic gasket seating width.
C	bolt circle diameter.
C_{ul}	conversion factor for length, $C_{ul} = 1.0$ for US Customary Units and $C_{ul} = 25.4$ for Metric Units.
C_{us}	conversion factor for stress, $C_{us} = 1.0$ for US Customary Units and $C_{us} = 6.894757E-03$ for Metric Units.
d	flange stress factor.
d_r	flange stress factor d for a reverse type flange.
E_{yg}	Modulus of Elasticity at the gasket seating load case temperature.
E_{yo}	Modulus of Elasticity at the operating load case temperature.
e	flange stress factor.
e_r	flange stress factor e for a reverse type flange.
F	flange stress factor for integral type flanges.
F_A	value of the external tensile net-section axial force. Compressive net-section forces are to be neglected and for that case, F_A should be taken as equal to zero.
F_L	flange stress factor for loose type flanges.
F_s	moment factor used to design split rings (see paragraph 4.16.8), $F_s = 1.0$ for non-split rings.
f	hub stress correction factor for integral flanges.
G	diameter at the location of the gasket load reaction (see Figure 4.16.8).
G_{avg}	average of the hub thicknesses g_1 and g_0.
G_c	outside diameter of the gasket contact area (see Figure 4.16.8).
g_0	thickness of the hub at the small end.

g_1	thickness of the hub at the large end.
H	total hydrostatic end force.
H_D	total hydrostatic end force on the area inside of the flange.
H_G	gasket load for the operating condition.
H_T	difference between the total hydrostatic end force and hydrostatic end force on the area inside the flange.
h	hub length.
h_o	hub length parameter.
h_{or}	hub length parameter for a reverse flange.
h_D	moment arm for load H_D.
h_G	moment arm for load H_G.
h_T	moment arm for load H_T.
I	bending moment of inertia of the flange cross-section.
I_p	polar moment of inertia of the flange cross-section.
J	flange rigidity index.
K	ratio of the flange outside diameter to the flange inside diameter.
K_R	rigidity index factor.
L	flange stress factor.
L_r	flange stress factor L for a reverse type flange.
M_E	absolute value of the external net-section bending moment.
M_g	flange design moment for the gasket seating condition.
M_o	flange design moment for the operating condition.
M_{oe}	component of the flange design moment resulting from a net section bending moment and/or axial force.
m	factor for the gasket operating condition.
N	gasket contact width, N = 0.0 for self-energizing gaskets.
P	design pressure.
r_1	radius to be at least $0.25g_1$ but not less than 5 mm (0.1875 in.).
r	radius of the undercut on a flange with nut stops.
S_{bg}	allowable stress from Annex 3.A for the bolt evaluated at the gasket seating temperature.
S_{bo}	allowable stress from Annex 3.A for the bolt evaluated at the design temperature.
S_{fg}	allowable stress from Annex 3.A for the flange evaluated at the gasket seating temperature.
S_{fo}	allowable stress from Annex 3.A for the flange evaluated at the design temperature.
S_{ng}	allowable stress from Annex 3.A for the nozzle neck, vessel, or pipe evaluated at the gasket seating temperature.
S_{no}	allowable stress from Annex 3.A for the nozzle neck, vessel, or pipe evaluated at the design temperature.
S_H	flange hub stress.
S_R	flange radial stress.

S_T	flange tangential stress.
S_{T1}	flange tangential stress at the outside diameter of a reverse flange.
S_{T2}	flange tangential stress at the inside diameter of a reverse flange.
T	flange stress factor.
T_r	flange stress factor T for a reverse flange.
t	flange thickness, including the facing thickness or the groove depth if either do not exceed 2 mm (0.0625 in.); otherwise, the facing thickness or groove depth is not included in the overall flange thickness.
t_n	nominal thickness of the shell, pipe, or nozzle to which the flange is attached.
t_x	is $2g_0$ when the design is calculated as an integral flange, or two times the minimum required thickness of the shell or nozzle wall when the design is based on a loose flange, but not less than 6 mm (0.25 in.).
U	flange stress factor.
U_r	flange stress factor U for a reverse type flange.
V	flange stress factor for integral type flanges.
V_L	flange stress factor for loose type flanges.
W_g	design bolt load for the gasket seating condition.
W_o	design bolt load for the operating condition.
w	width of the nubbin.
y	factor for the gasket seating condition
Y	flange stress factor.
Y_r	flange stress factor Y for a reverse type flange.
Z	flange stress factor.

4.16.13 Tables

Table 4.16.1 – Gasket Factors For Determining The Bolt Loads

Gasket Material	Gasket Factor m	Min. Design Seating Stress y, MPa (psi)	Column in Table 4.16.3	Facing Sketch In Table 4.16.3
Self-energizing types (O rings, metallic, elastomer, other gasket types considered as self-sealing)	0	0	---	---
Elastomers without fabric or high percent of mineral fiber: • below 75 A Shore Durometer • 75 A or higher Shore Durometer	 0.50 1.00	 0 1.4 (200)	II	(1a), (1b), (1c), (1d), (4), (5)
Mineral fiber with suitable binder for operating conditions: • 3.2 mm (1/8 inch) thick • 1.6 mm (1/16 inch) thick • 0.8 mm (1/32 inch) thick	 2.00 2.75 3.50	 11 (1,600) 26 (3,700) 45 (6,500)	II	(1), (1b), (1c), (1d), (4), (5)
Elastomers with cotton fabric insertion	1.25	2.8 (400)	II	(1a), (1b), (1c), (1d), (4), (5)
Elastomers with mineral fiber insertion (with or without wire reinforcement): • 3-ply • 2-ply • 1-ply	 2.25 2.50 2.75	 15 (2200) 20 (2,900) 26 (3,700)	II	(1), (1b), (1c), (1d), (5)
Vegetable fiber	1.75	7.6 (1,100)	II	(1a), (1b), (1c), (1d), (4), (5)
Spiral-wound metal, mineral fiber filler • Carbon steel • Stainless steel, Monel, and nickel-base alloy	 2.50 3.00	 69 (10,000) 69 (10,000)	II	(1a), (1b)
Corrugated metal, mineral fiber inserted, or corrugated metal, jacketed mineral fiber filled: • Soft aluminum • Soft copper or brass • Iron or soft steel • Monel or 4% - 6% chrome • Stainless steels and nickel-base alloys	 2.50 2.75 3.00 3.25 3.50	 20 (2,900) 26 (3,700) 31 (4,500) 38 (5,500) 45 (6,500)	II	(1a), (1b)
Corrugated metal: • Soft aluminum • Soft copper or brass • Iron or soft steel • Monel or 4% - 6% chrome • Stainless steels and nickel-base alloys	 2.75 3.00 3.25 3.50 3.75	 26 (3,700) 31 (4,500) 38 (5,500) 45 (6,500) 52 (7,600)	II	(1a), (1b), (1c), (1d)

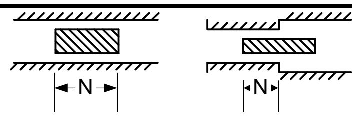
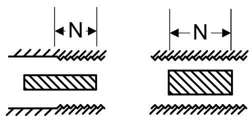
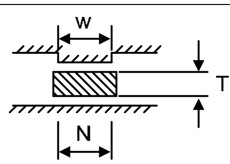
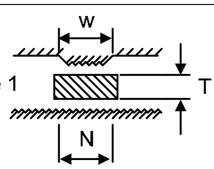
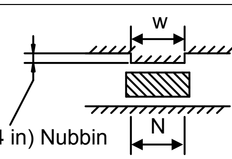
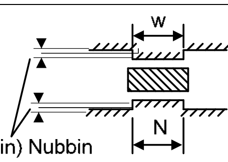
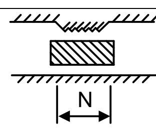
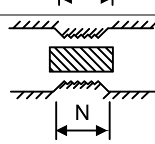
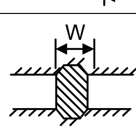
Table 4.16.1 – Gasket Factors For Determining The Bolt Loads

Gasket Material	Gasket Factor m	Min. Design Seating Stress y, MPa (psi)	Column in Table 4.16.3	Facing Sketch In Table 4.16.3
Flat metal, jacketed mineral fiber filled: • Soft aluminum • Soft copper or brass • Iron or soft steel • Monel • 4% - 6% chrome • Stainless steels and nickel-base alloys	 3.25 3.50 3.75 3.50 3.75 3.75	 38 (5,500) 45 (6,500) 52 (7,600) 55 (8,000) 62 (9,000) 62 (9,000)	II	(1a), (1b), (1c), (1d), (2)
Grooved Metal: • Soft aluminum • Soft copper or brass • Iron or soft steel • Monel or 4% - 6% chrome • Stainless steels and nickel-base alloys	 3.25 3.50 3.75 3.75 4.25	 38 (5,500) 45 (6,500) 52 (7,600) 62 (9,000) 70 (10,100)	II	(1a), (1b), (1c), (1d), (2), (3)
Sold flat metal: • Soft aluminum • Soft copper or brass • Iron or soft steel • Monel or 4% - 6% chrome • Stainless steels and nickel-base alloys	 4.00 4.75 5.50 6.00 6.50	 61 (8,800) 90 (13,000) 124 (18,000) 150 (21,800) 180 (26,000)	I	(1a), (1b), (1c), (1d), (2), (3), (4), (5)
Ring joint: • Iron or soft steel • Monel or 4% - 6% chrome • Stainless steel and nickel-base alloys	 5.50 6.00 6.50	 124 (18,000) 150 (21,800) 180 (26,000)	I	(6)
Note: This table gives a list of commonly used gasket materials and contact facings with suggested values of m and y that have generally proved satisfactory in actual service when using effective gasket seating width b. The design values and other details given in this table are suggested only and are not mandatory.				

Table 4.16.2 – Recommended Minimum Gasket Contact Width

Gasket Contact Width, N					
Gasket Type	Gasket Outside Diameter				
	< 150 mm (6 inch)	< 300 mm (12 inch)	< 600 mm (24 inch)	< 900 mm (36 inch)	900 mm (36 inch) and Over
Sheet Gaskets Including Laminated Sheets Gaskets With Or Without A Metal Core	9 mm (0.375in)	12 mm (0.5in)	16 mm (0.625in)	16 mm (0.625in)	19 mm (0.75in)
Preformed Composite Gaskets Including Spiral Wound, Jacketed, And Solid Flat Metal Gaskets	6 mm (0.25in)	9 mm (0.375in)	12 mm (0.5in)	16 mm (0.625in)	16 mm (0.625in)

Table 4.16.3 – Effective Gasket Width For Determining The Bolt Loads

Facing Sketch	Facing Sketch Detail (Exaggerated)	Basic Gasket Seating Width b_o	
		Column I	Column II
1a		$\frac{N}{2}$	$\frac{N}{2}$
1b	See Note 1 		
1c	 w ≤N	$\min\left[\frac{w+T}{2}, \frac{w+N}{4}\right]$	$\min\left[\frac{w+T}{2}, \frac{w+N}{4}\right]$
1d	See Note 1  w ≤ N		
2	 w ≤ N/2 0.4 mm (1/64 in) Nubbin	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
3	 w ≤ N/2 0.4 mm (1/64 in) Nubbin	$\frac{N}{4}$	$\frac{3N}{8}$
4	See Note 1 	$\frac{3N}{8}$	$\frac{7N}{16}$
5	See Note 1 	$\frac{N}{4}$	$\frac{3N}{8}$
6		$\frac{w}{8}$	– – –

Notes:
1. Where serrations do not exceed 0.4 mm, (0.0156in) depth and 0.8mm (0.0313in) width spacing, Sketches (1b) and (1d) shall be used.
2. The gasket factors listed in this table only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

Table 4.16.4 – Flange Stress Factors Equations Involving Diameter

Flange Type	Stress Factors Involving Diameter
Integral Type Flange and Loose Type Flange with a Hub	$K = \frac{A}{B}$ $Y = \frac{1}{K-1}\left[0.66845 + 5.71690\left(\frac{K^2 \log_{10} K}{K^2 - 1}\right)\right]$ $T = \frac{K^2\left(1 + 8.55246 \log_{10} K\right) - 1}{\left(1.04720 + 1.9448 K^2\right)\left(K - 1\right)}$ $U = \frac{K^2\left(1 + 8.55246 \log_{10} K\right) - 1}{1.36136\left(K^2 - 1\right)\left(K - 1\right)}$ $Z = \frac{\left(K^2 + 1\right)}{\left(K^2 - 1\right)}$ $L = \frac{te + 1}{T} + \frac{t^3}{d}$ $e = \frac{F}{h_o}$ *for Integral Type Flanges* $e = \frac{F_L}{h_o}$ *for Loose Type Flanges with a Hub* $d = \frac{U g_o^2 h_o}{V}$ *for Integral Type Flanges* $d = \frac{U g_o^2 h_o}{V_L}$ *for Loose Type Flanges with a Hub* $h_o = \sqrt{B g_0}$ $X_g = \frac{g_1}{g_0}$ $X_h = \frac{h}{h_o}$

Table 4.16.4 – Flange Stress Factors Equations Involving Diameter

<table>
<tr><th>Flange Type</th><th>Stress Factors Involving Diameter</th></tr>
<tr><td>Reverse Integral Type Flange and Reverse Loose Type Flanges with a Hub</td><td>
The parameters K, T, U, Y, and Z are determined using the equations for Integral and Loose Type Flanges with:

$$K = \frac{A}{B^*}$$

Then, the reverse flange parameters are computed as follows:

$$Y_r = \alpha_r Y$$

$$T_r = \frac{(Z+0.3)}{(Z-0.3)}\alpha_r T$$

$$U_r = \alpha_r U$$

$$L_r = \frac{te_r + 1}{T_r} + \frac{t^3}{d_r}$$

$$\alpha_r = \frac{1}{K^2}\left[1 + \frac{0.668(K+1)}{Y}\right]$$

$$e_r = \frac{F}{h_{or}} \qquad \textit{for Integral Type Flanges}$$

$$e_r = \frac{F_L}{h_{or}} \qquad \textit{for Loose Type Flanges with a Hub}$$

$$d_r = \frac{U_r g_o^2 h_{or}}{V} \qquad \textit{for Integral Type Flanges}$$

$$d_r = \frac{U_r g_o^2 h_{or}}{V_L} \qquad \textit{for Loose Type Flanges with a Hub}$$

$$h_{or} = \sqrt{A g_0}$$

$$X_g = \frac{g_1}{g_0}$$

$$X_h = \frac{h}{h_{or}}$$
</td></tr>
</table>

Table 4.16.5 – Flange Stress Factor Equations

Flange Type	Stress Factors
Integral Type Flange, Reverse Integral Type Flange	$F = \left(\begin{array}{l} 0.897697 - 0.297012 \ln X_g + 9.5257\left(10^{-3}\right) \ln X_h + \\ 0.123586\left(\ln X_g\right)^2 + 0.0358580\left(\ln X_h\right)^2 - 0.194422\left(\ln X_g\right)\left(\ln X_h\right) - \\ 0.0181259\left(\ln X_g\right)^3 + 0.0129360\left(\ln X_h\right)^3 - \\ 0.0377693\left(\ln X_g\right)\left(\ln X_h\right)^2 + 0.0273791\left(\ln X_g\right)^2\left(\ln X_h\right) \end{array} \right)$ $For\ \ 0.1 \le X_h \le 0.5$ $V = \left(\begin{array}{l} 0.500244 + \dfrac{0.227914}{X_g} - 1.87071 X_h - \dfrac{0.344410}{X_g^2} + 2.49189 X_h^2 + \\ 0.873446\left(\dfrac{X_h}{X_g}\right) + \dfrac{0.189953}{X_g^3} - 1.06082 X_h^3 - 1.49970\left(\dfrac{X_h^2}{X_g}\right) + \\ 0.719413\left(\dfrac{X_h}{X_g^2}\right) \end{array} \right)$ $For\ \ 0.5 < X_h \le 2.0$ $V = \left(\begin{array}{l} 0.0144868 - \dfrac{0.135977}{X_g} - \dfrac{0.0461919}{X_h} + \dfrac{0.560718}{X_g^2} + \dfrac{0.0529829}{X_h^2} + \\ \dfrac{0.244313}{X_g X_h} + \dfrac{0.113929}{X_g^3} - \dfrac{0.00928265}{X_h^3} - \dfrac{0.0266293}{X_g X_h^2} - \dfrac{0.217008}{X_g^2 X_h} \end{array} \right)$ $f = \max\left[1.0,\ \left(\dfrac{\begin{array}{l} 0.0927779 - 0.0336633 X_g + 0.964176 X_g^2 + \\ 0.0566286 X_h + 0.347074 X_h^2 - 4.18699 X_h^3 \end{array}}{\begin{array}{l} 1 - 5.96093(10^{-3}) X_g + 1.62904 X_h + \\ 3.49329 X_h^2 + 1.39052 X_h^3 \end{array}} \right) \right]$

Table 4.16.5 – Flange Stress Factor Equations

Flange Type	Stress Factors
Loose Type Flange with a Hub, Reverse Loose Type Flange with a Hub	$F_L = \left(\dfrac{0.941074 + 0.176139\left(\ln X_g\right) - 0.188556\left(\ln X_h\right) + 0.0689847\left(\ln X_g\right)^2 + 0.523798\left(\ln X_h\right)^2 - 0.513894\left(\ln X_g\right)\left(\ln X_h\right)}{1 + 0.379392\left(\ln X_g\right) + 0.184520\left(\ln X_h\right) - 0.00605208\left(\ln X_g\right)^2 - 0.00358934\left(\ln X_h\right)^2 + 0.110179\left(\ln X_g\right)\left(\ln X_h\right)} \right)$ $For \;\; 0.1 \le X_h \le 0.25$ $\ln\left[V_L\right] = \left(6.57683 - 0.115516 X_g + 1.39499\sqrt{X_g}\left(\ln X_g\right) + 0.307340\left(\ln X_g\right)^2 - 8.30849\sqrt{X_g} + 2.62307\left(\ln X_g\right) + 0.239498 X_h\left(\ln X_h\right) - 2.96125\left(\ln X_h\right) + \dfrac{7.035052(10^{-4})}{X_h} \right)$ $For \;\; 0.25 < X_h \le 0.50$ $V_L = \left(1.56323 - 1.80696\left(\ln X_g\right) - \dfrac{1.33458}{X_h} + 0.276415\left(\ln X_g\right)^2 + \dfrac{0.417135}{X_h^2} + \dfrac{1.39511\left(\ln X_g\right)}{X_h} + 0.0137129\left(\ln X_g\right)^3 + \dfrac{0.0943597}{X_h^3} - \dfrac{0.402096\left(\ln X_g\right)}{X_h^2} - \dfrac{0.101619\left(\ln X_g\right)^2}{X_h} \right)$ $For \;\; 0.50 < X_h \le 1.0$ $V_L = \left(-0.0213643 - \dfrac{0.0763597}{X_g} + \dfrac{0.102990}{X_h} + \dfrac{0.725776}{X_g^2} - \dfrac{0.160603}{X_h^2} - \dfrac{0.0918061}{X_g \cdot X_h} + \dfrac{0.472277}{X_g^3} + \dfrac{0.0873530}{X_h^3} + \dfrac{0.527487}{X_g \cdot X_h^2} - \dfrac{0.980209}{X_g^2 \cdot X_h} \right)$ $For \;\; 1.0 < X_h \le 2.0$ $V_L = \left(7.96687(10^{-3}) - \dfrac{0.220518}{X_g} + \dfrac{0.0602652}{X_h} + \dfrac{0.619818}{X_g^2} - \dfrac{0.223212}{X_h^2} + \dfrac{0.421920}{X_g \cdot X_h} + \dfrac{0.0950195}{X_g^3} + \dfrac{0.209813}{X_h^3} - \dfrac{0.158821}{X_g \cdot X_h^2} - \dfrac{0.242056}{X_g^2 \cdot X_h} \right)$ $f = 1.0$

Table 4.16.6 – Moment Arms For Flange Loads For The Operating Condition

Flange Type	h_D	h_T	h_G
Integral Type Flanges	$\frac{C-B-g_1}{2}$	$\frac{1}{2}\left[\frac{C-B}{2}+h_G\right]$	$\frac{C-G}{2}$
Loose Type Flanges, except Lap Joint Flanges	$\frac{C-B}{2}$	$\frac{h_D+h_G}{2}$	$\frac{C-G}{2}$
Loose Type Lap Joint Flanges	$\frac{C-B}{2}$	$\frac{C-G}{2}$	$\frac{C-G}{2}$
Reverse Integral Type Flanges	$\frac{C+g_1-2g_o-B}{2}$	$\frac{1}{2}\left(C-\frac{B+G}{2}\right)$	$\frac{C-G}{2}$
Reverse Loose Type Flanges	$\frac{C-B}{2}$	$\frac{1}{2}\left(C-\frac{B+G}{2}\right)$	$\frac{C-G}{2}$

Table 4.16.7 – Flange Moments Of Inertia

<table>
<tr><th>Flange Type</th><th>I</th><th>I_p</th></tr>
<tr><td>Integral Type Flange with a Hub</td><td>$I = \dfrac{0.0874 L g_o^2 h_o B}{V}$</td><td rowspan="2">$I_p = K_{AB} + K_{CD}$
$K_{AB} = \left(A_A B_B^3\right)\left[\dfrac{1}{3} - 0.21\left(\dfrac{B_B}{A_A}\right)\left(1 - \dfrac{1}{12}\left\{\dfrac{B_B}{A_A}\right\}^4\right)\right]$
$K_{CD} = \left(C_C D_{DG}^3\right)\left[\dfrac{1}{3} - 0.105\left(\dfrac{D_{DG}}{C_C}\right)\left(1 - \dfrac{1}{192}\left\{\dfrac{D_{DG}}{C_C}\right\}^4\right)\right]$
$A_R = 0.5\left(A - B\right)$
$G_{avg} = 0.5\left(g_0 + g_1\right)$

$If\ \ t \ge G_{avg}:$
$A_A = A_R,\ \ B_B = t,\ \ C_C = h,\ \ D_{DG} = G_{avg}$
$If\ \ t < G_{avg}:$
$A_A = h + t,\ \ B_B = G_{avg},\ \ C_C = A_R - G_{avg},\ \ D_{DG} = t$</td></tr>
<tr><td>Loose Type Flange with a Hub</td><td>$I = \dfrac{0.0874 L g_o^2 h_o B}{V_L}$</td></tr>
<tr><td>Loose Type Flange without a Hub</td><td>$I = \dfrac{B t^3 \ln K}{6}$</td><td>$I_p = A_R t^3\left[\dfrac{1}{3} - 0.21\left(\dfrac{t}{A_R}\right)\left(1 - \dfrac{1}{12}\left\{\dfrac{t}{A_R}\right\}^4\right)\right]$
$A_R = 0.5\left(A - B\right)$</td></tr>
</table>

Table 4.16.8 – Flange Stress Equations

Flange Type	Stress Equations	
	Operating Condition	**Gasket Seating Conditions**
Integral Type Flange or Loose Type Flange with a Hub	$S_H = \frac{fM_o}{Lg_1^2 B}$ $S_R = \frac{(1.33te+1)M_o}{Lt^2 B}$ $S_T = \frac{YM_o}{t^2 B} - ZS_R$	$S_H = \frac{fM_g}{Lg_1^2 B}$ $S_R = \frac{(1.33te+1)M_g}{Lt^2 B}$ $S_T = \frac{YM_g}{t^2 B} - ZS_R$
Loose Type Flange without a Hub	$S_T = \frac{YM_o}{t^2 B}$	$S_T = \frac{YM_g}{t^2 B}$
Reverse Integral Type Flange or Reverse Loose Type Flange with a Hub	$S_H = \frac{fM_o}{L_r g_1^2 B^*}$ $S_R = \frac{(1.33te_r+1)M_o}{L_r t^2 B^*}$ $S_{T1} = \frac{Y_r M_o}{t^2 B^*} - \frac{ZS_R(0.67te_r+1)}{(1.33te_r+1)}$ $S_{T2} = \left[Y - \frac{2K^2(0.67te_r+1)}{(K^2-1)L_r}\right]\frac{M_o}{t^2 B^*}$	$S_H = \frac{fM_g}{L_r g_1^2 B^*}$ $S_R = \frac{(1.33te_r+1)M_g}{L_r t^2 B^*}$ $S_{T1} = \frac{Y_r M_g}{t^2 B^*} - \frac{ZS_R(0.67te_r+1)}{(1.33te_r+1)}$ $S_{T2} = \left[Y - \frac{2K^2(0.67te_r+1)}{(K^2-1)L_r}\right]\frac{M_g}{t^2 B^*}$
Reverse Loose Type Flange without a hub	$S_T = \frac{YM_o}{t^2 B^*}$	$S_T = \frac{YM_g}{t^2 B^*}$

Table 4.16.9 – Flange Stress Acceptance Criteria

Flange Type	Stress Acceptance Criteria	
	Operating Condition	**Gasket Seating Conditions**
Integral Type Flange or Loose Type Flange with a Hub	$S_H \le \min\left[1.5S_{fo},\ 2.5S_{no}\right]$ (1) $S_H \le 1.5S_{fo}$ (2) $S_R \le S_{fo}$ $S_T \le S_{fo}$ $\frac{(S_H + S_R)}{2} \le S_{fo}$ $\frac{(S_H + S_T)}{2} \le S_{fo}$	$S_H \le \min\left[1.5S_{fg},\ 2.5S_{ng}\right]$ (1) $S_H \le 1.5S_{fg}$ (2) $S_R \le S_{fg}$ $S_T \le S_{fg}$ $\frac{(S_H + S_R)}{2} \le S_{fg}$ $\frac{(S_H + S_T)}{2} \le S_{fg}$
Loose Type Flange without a Hub	$S_T \le S_{fo}$	$S_T \le S_{fg}$
Reverse Integral Type Flange or Reverse Loose Type Flange with a Hub	$S_H \le 1.5S_{fo}$ $S_R \le S_{fo}$ $S_{T1} \le S_{fo}$ $\frac{(S_H + S_R)}{2} \le S_{fo}$ $\frac{(S_H + S_{T1})}{2} \le S_{fo}$ $S_{T2} \le S_{fo}$	$S_H \le 1.5S_{fg}$ $S_R \le S_{fg}$ $S_{T1} \le S_{fg}$ $\frac{(S_H + S_R)}{2} \le S_{fg}$ $\frac{(S_H + S_{T1})}{2} \le S_{fg}$ $S_{T2} \le S_{fg}$
Reverse Loose Type Flanges	$S_T \le S_{fo}$	$S_T \le S_{fg}$

Notes:
1. For integral flanges with hubs welded to a nozzle neck, pipe, or vessel shell
2. For loose type flanges with a hub.
3. Flanges made of non-ductile material, such as cast iron, are not addressed by this section.

Table 4.16.10 – Flange Rigidity Criterion

Flange Type	Rigidity Criterion	
	Operating Condition	**Gasket Seating Conditions**
Integral Type Flange	$J=\frac{52.14VM_o}{LE_{yo}g_0^2K_Rh_o}\le 1.0$	$J=\frac{52.14VM_g}{LE_{yg}g_0^2K_Rh_o}\le 1.0$
Loose Type Flange with a Hub	$J=\frac{52.14V_LM_o}{LE_{yo}g_0^2K_Rh_o}\le 1.0$	$J=\frac{52.14V_LM_g}{LE_{yg}g_0^2K_Rh_o}\le 1.0$
Reverse Integral Type Flange	$J=\frac{52.14VM_o}{L_rE_{yo}g_0^2K_Rh_o}\le 1.0$	$J=\frac{52.14VM_g}{L_rE_{yg}g_0^2K_Rh_o}\le 1.0$
Reverse Loose Type Flange with a Hub	$J=\frac{52.14V_LM_o}{L_rE_{yo}g_0^2K_Rh_o}\le 1.0$	$J=\frac{52.14V_LM_g}{L_rE_{yg}g_0^2K_Rh_o}\le 1.0$
Loose Type and Reverse Loose Type Flange without a Hub	$J=\frac{109.4M_o}{E_{yo}t^3K_R\left(\ln K\right)}\le 1.0$	$J=\frac{109.4M_g}{E_{yg}t^3K_R\left(\ln K\right)}\le 1.0$

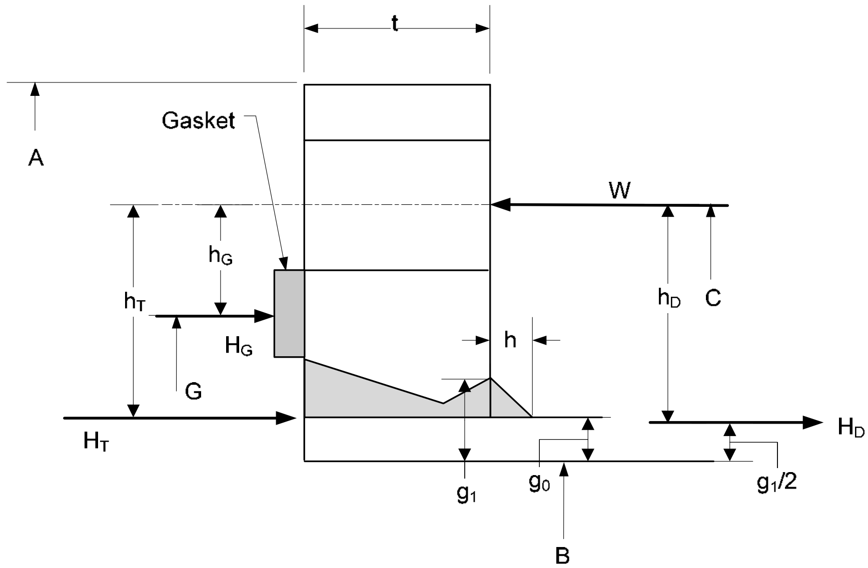
Notes:

1. For an integral type flange, $K_R=0.3$ unless other values are specified by the user.
2. For a loose type flange with or without a hub, $K_R=0.2$ unless other values are specified by the user.

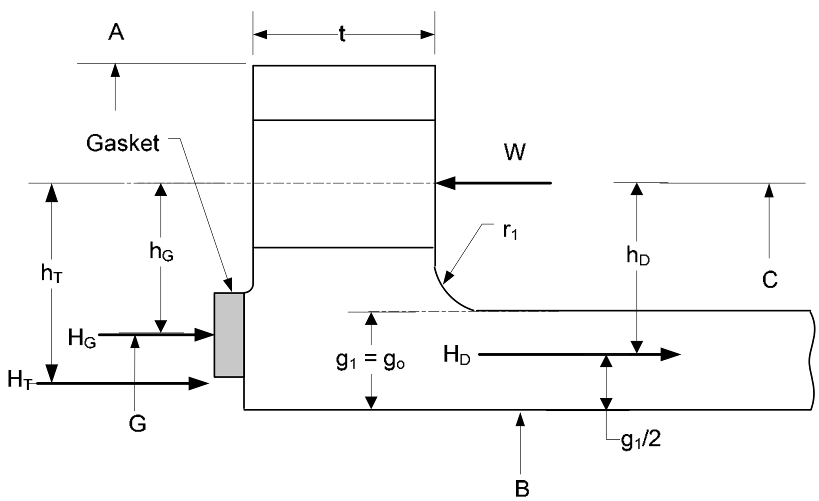
Table 4.16.11 - Bolt Spacing Equations

Flange Type	Bolt Spacing Factors
All	$B_{s\max}=2a+\frac{6t}{m+0.5}$
	$B_{sc}=\sqrt{\frac{B_s}{2a+t}}$

4.16.14 Figures

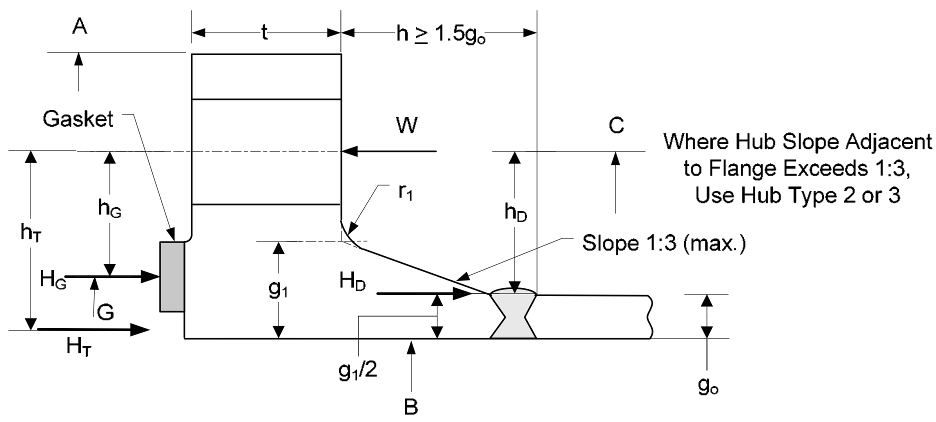


(a) Integral Flange Without A Hub

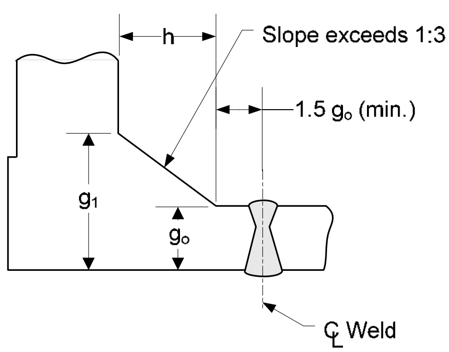


(b) Integral Flange with g_o=g_1

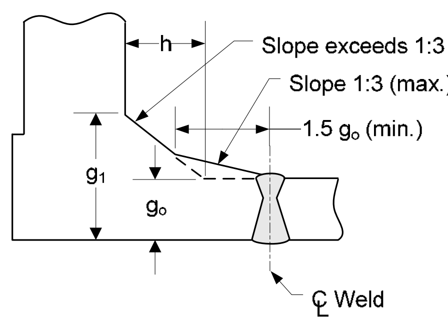
Figure 4.16.1 – Integral Type Flanges



(a) Hub Type 1

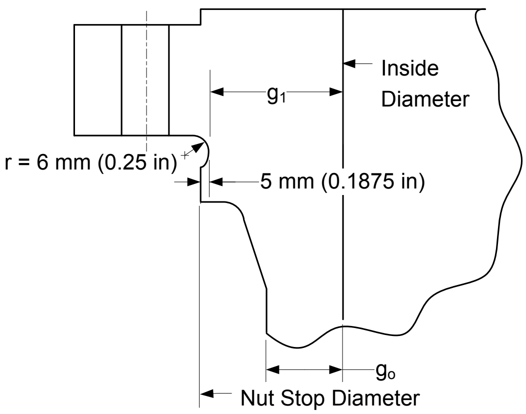


(b) Hub Type 2

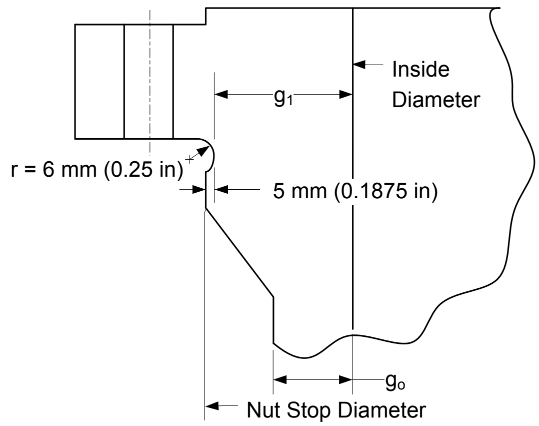


(c) Hub Type 3

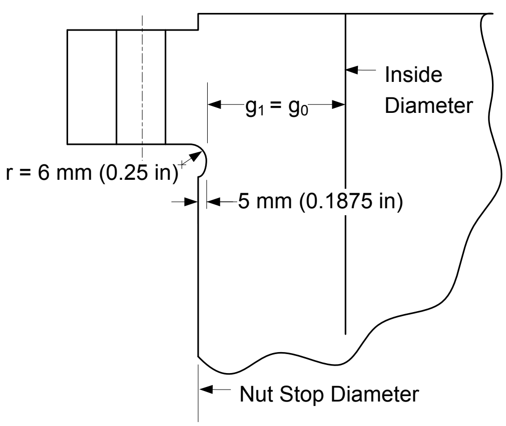
Figure 4.16.2 – Integral Type Flanges with a Hub



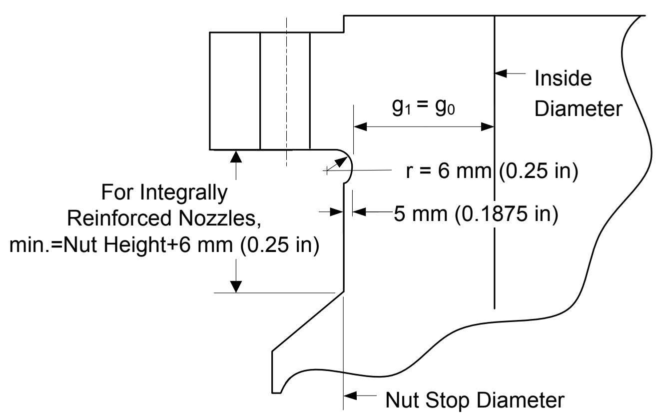
(a) Detail A



(b) Detail B

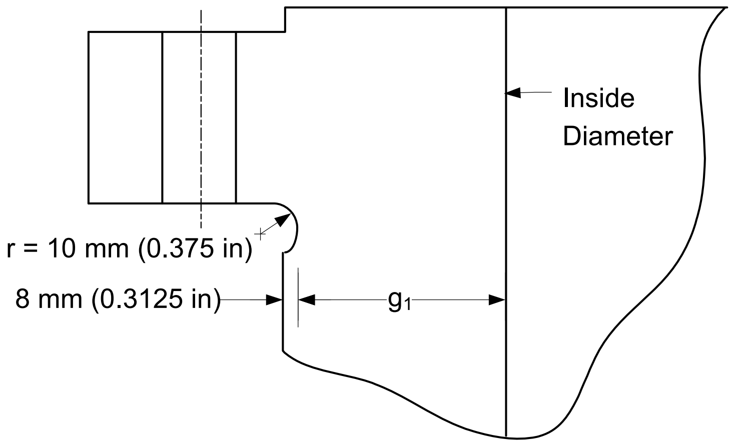


(c) Detail C



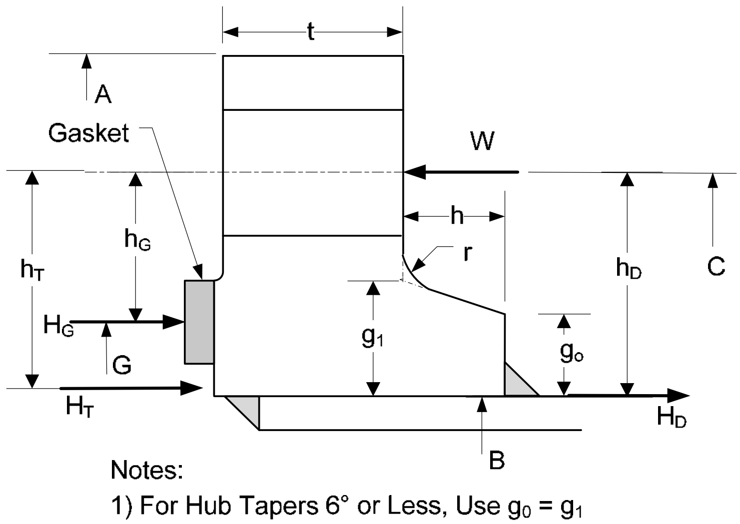
(d) Detail D

Figure 4.16.3 – Integral Type Flanges With Nut Stops – Diameter Less Than or Equal To 450mm (18 Inches)

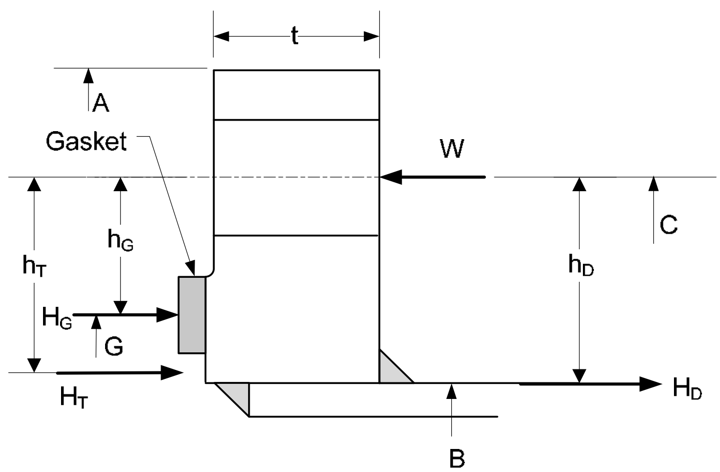


Note: All other details per Figure 4.16.3

Figure 4.16.4 – Integral Type Flanges With Nut Stops – Diameter Greater Than 450mm (18 Inches)

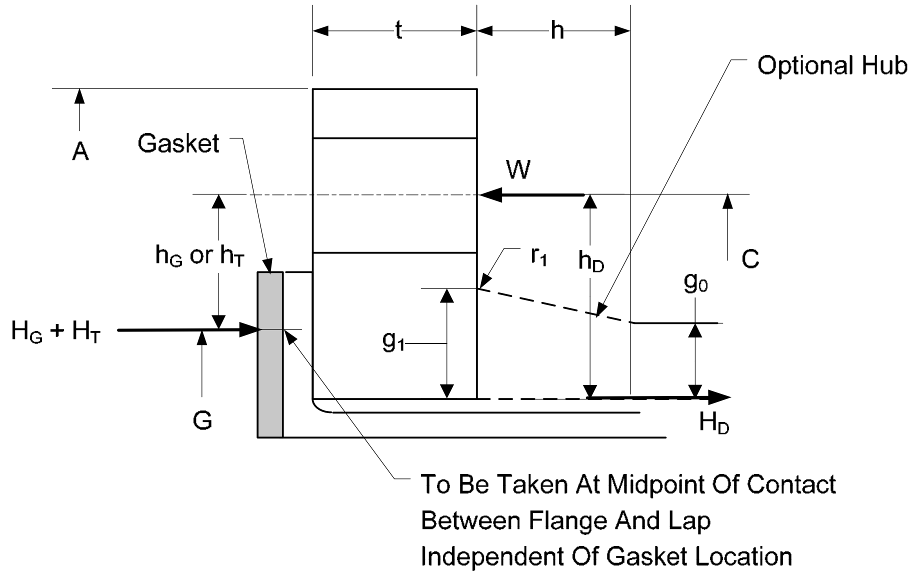


(a) Loose Flange With A Hub



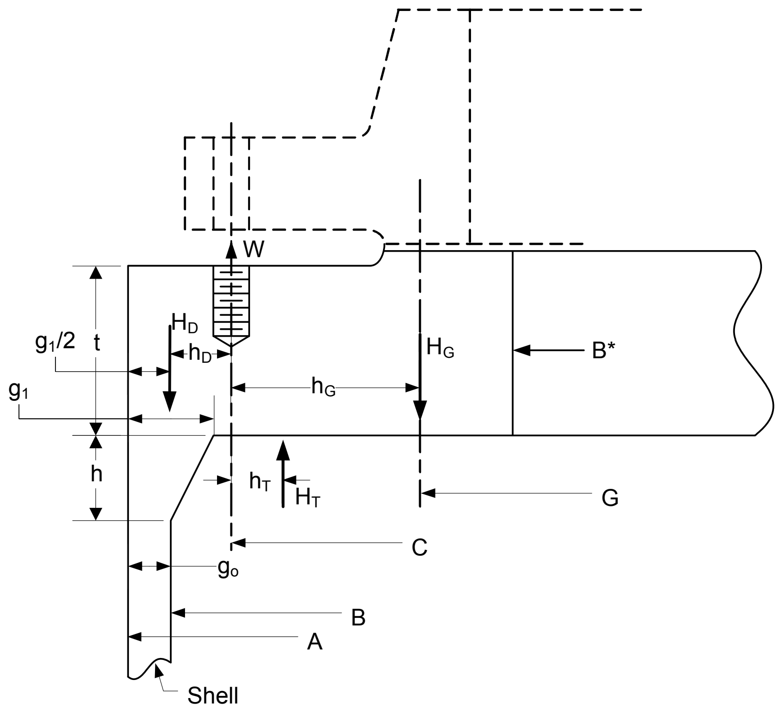
(b) Loose Flange Without A Hub

Figure 4.16.5 – Loose Type Flanges

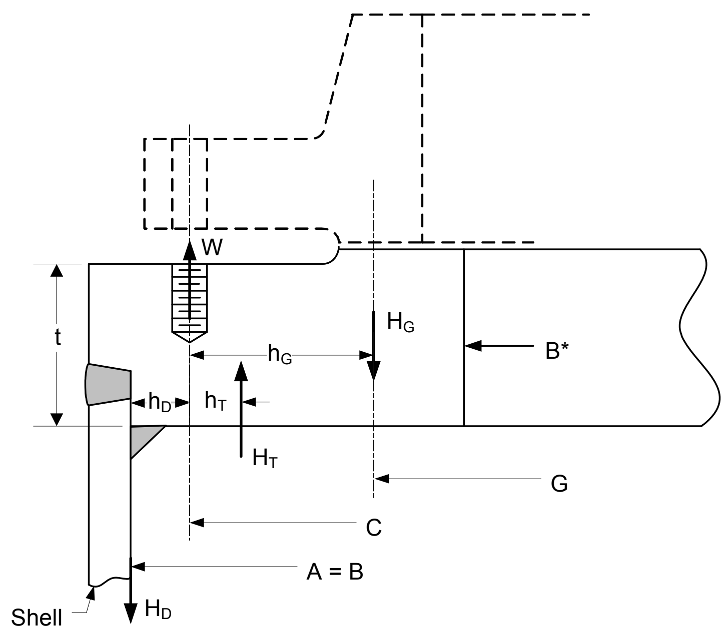


Notes (Loose Type Flanges):
(1) For Hub Tapers 6° or Less, Use $g_0 = g_1$

Figure 4.16.6 – Loose Type Lap Joint Type Flanges



(a) Integral Type Reverse Flange



(b) Loose Type Reverse Flange

Figure 4.16.7 – Reverse Flanges

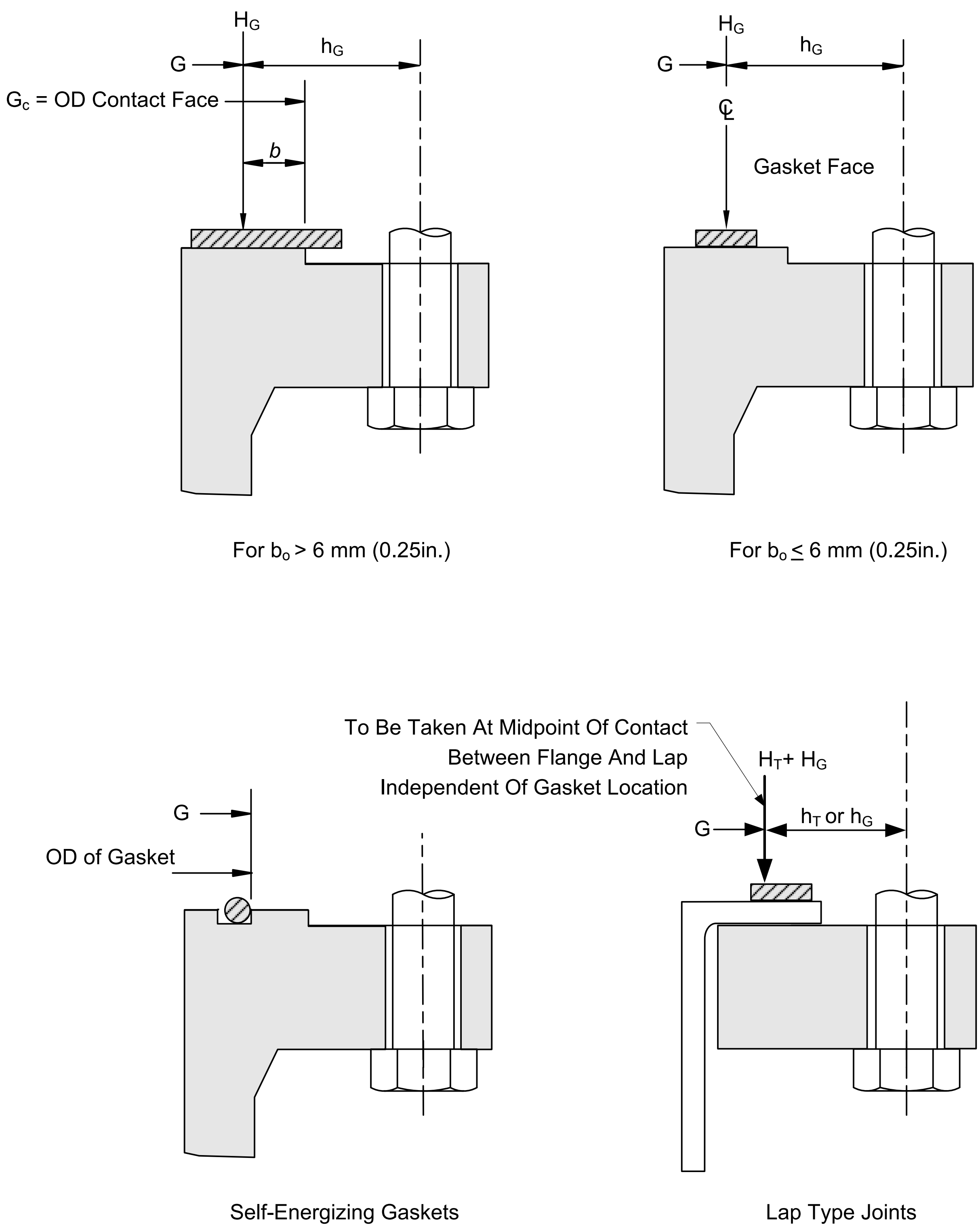


Figure 4.16.8 – Location of Gasket Reaction Load Diameter

4.17 Design Rules for Clamped Connections

4.17.1 Scope

The rules in paragraph 4.17 apply specifically to the design of clamp connections for pressure vessels and vessel parts. These rules shall not be used for the determination of thickness of supported or unsupported tubesheets integral with a hub nor for the determination of thickness of covers. These rules provide only for hydrostatic end loads, assembly, and gasket seating.

4.17.2 Design Considerations

4.17.2.1 The design of a clamp connection involves the selection of the gasket, bolting, hub, and clamp geometry. Connection dimensions shall be such that the calculated stresses in the clamp and the hub do not exceed the acceptability criteria of this paragraph.

4.17.2.2 In the design of a bolted flange connection, calculations shall be made for the following two design conditions, and the most severe condition shall govern the design of the flanged joint.

a) *Operating Conditions* – The conditions required to resist the hydrostatic end force of the design pressure and any applied external forces and moments tending to part the joint, and to maintain on the gasket or joint-contact surface sufficient compression to assure a joint that meets the required tightness, all at the design temperature.

b) *Gasket Seating And Assembly Condition* – The conditions existing when the gasket or joint-contact surface is seated by applying an initial load with the bolts when assembling the joint, at atmospheric temperature and pressure.

4.17.2.3 Calculations shall be performed using dimensions of the flange in the corroded condition and the uncorroded condition, and the more severe case shall control.

4.17.2.4 It is recommended that either a pressure energized and/or low seating load gasket be used to compensate for possible non-uniformity in the gasket seating force distribution. Hub faces shall be designed such as to have metal-to-metal contact outside the gasket seal diameter. This may be provided by recessing the hub faces or by use of a metal spacer (see Figure 4.17.1). The contact area shall be sufficient to prevent yielding of either the hub face or spacer under both operating and assembly loads.

4.17.2.5 It is recognized that there are clamp designs that do not utilize wedging action during assembly since clamping surfaces are parallel to the hub faces. Such designs are acceptable and shall satisfy the bolting and corresponding clamp and hub requirements of a clamp connection designed for a total included clamping angle of 10 degrees.

4.17.2.6 The design method used in this paragraph to calculate stresses, loads, and moments may also be used for designing clamp connections of shapes differing from those shown in Figures 4.17.1 and 4.17.2, and for clamps consisting of more than two circumferential segments. The design equations in this paragraph may be modified when designing clamp connections of shapes differing from those shown in Figures 4.17.1 and 4.17.2, provided that the basis for the modifications is in accordance with paragraph 1.1.1.2. The clamp connections designed in this manner shall be provided with a bolt retainer. The retainer shall be designed to hold the clamps together independently in case of failure of one of the primary bolting ([see paragraph 4.8). Multiple bolting (two or more bolts per lug) is an acceptable alternative for meeting this requirement. Clamp-hub friction shall not be considered as a retainer method.

4.17.3 Flange Materials

4.17.3.1 Materials used in the construction of clamp connections shall comply with the requirements given in Part 3.

4.17.3.2 Hubs made from ferritic steel and designed in accordance with the rules herein shall be given a normalizing or full-annealing heat treatment when the thickness of the hub neck section exceeds 76 mm (3 in.).

4.17.3.3 Cast steel hubs and clamps shall be examined and repaired, if required, in accordance with Part 3.

4.17.3.4 Hubs and clamps shall not be machined from plate.

4.17.3.5 Bolts, studs, nuts and washers shall comply with Part 3. The minimum bolt diameter shall be 13 mm (0.5 in.).

4.17.4 Design Bolt Loads

4.17.4.1 During assembly of the clamp connection, the design bolt load W is resolved into an effective clamp preload W_e, which is a function of the clamp-to-hub taper angle ϕ and the friction angle μ. An appropriate friction angle shall be established by the manufacturer, based on test results for both assembly and operating conditions

4.17.4.2 The procedure to determine the bolt loads for the operating, and gasket seating and assembly conditions are shown below.

a) STEP 1 – Determine the design pressure and temperature of the flange joint.

b) STEP 2 – Select a gasket and determine the gasket factors m and y from Table 4.16.1.

c) STEP 3 – Determine the width of the gasket, basic gasket seating width, the location of the gasket reaction, G, based on the flange and gasket geometry and the information in Table 4.16.3.

   1) When $b_0 \le 6\,mm\ (0.25\ in.)$, then G is the mean diameter of the gasket or joint contact face

   2) When $b_0 > 6\,mm\ (0.25\ in.)$, then G is the outside diameter of the gasket contact face minus $2b$.

d) STEP 4 – Determine the flange forces for the bolt load calculation.

$$H = 0.785G^2P \tag{4.17.1}$$

$$H_p = 2b\pi GmP \qquad for\ non-self-energized\ gaskets \tag{4.17.2}$$

$$H_m = \pi bGy \qquad for\ non-self-energized\ gaskets \tag{4.17.3}$$

$$H_p = 0.0 \qquad for\ self-energized\ gaskets \tag{4.17.4}$$

$$H_m = 0.0 \qquad for\ self-energized\ gaskets \tag{4.17.5}$$

Note that where a significant axial force is required to compress the gasket during assembly of a joint containing a self-energizing gasket, the value of H_m shall be taken as equal to that axial force.

e) STEP 5 – Determine the design bolt load for the operating condition.

$$W_o = \frac{2}{\pi}\left(H + H_p\right)\tan\left[\phi - \mu\right] \tag{4.17.6}$$

f) STEP 6 – Determine the minimum required total bolt load for gasket seating and assembly conditions.

$$W_{g1} = \frac{2}{\pi} H_m \tan\left[\phi + \mu\right] \tag{4.17.7}$$

$$W_{g2} = \frac{2}{\pi}\left(H + H_p\right)\tan\left[\phi + \mu\right] \tag{4.17.8}$$

g) STEP 7 – Determine the design bolt load for the gasket seating and assembly condition.

$$W_g = \frac{\left(A_m + A_b\right)S_{bg}}{2} \tag{4.17.9}$$

The parameter A_b is the actual total cross sectional area of the bolts that is selected such that $A_b \geq A_m$, where

$$A_m = \max\left[\frac{W_o}{2S_{bo}},\ \frac{W_{g1}}{2S_{bg}},\ \frac{W_{g2}}{2S_{bg}}\right] \tag{4.17.10}$$

4.17.4.3 In Equation (4.17.6), credit for friction is allowed based on clamp connection geometry and experience, but the bolt load shall not be less than that determined using a value of $\left(\phi - \mu\right)$ equal to 5°. Friction is also considered in determining bolt loads by Equation (4.17.7) and Equation (4.17.8), but the μ factor used shall not be less than 5°.

4.17.5 Flange and Clamp Design Procedure

The procedure to design a clamp connection is shown below.

a) STEP 1 – Determine the design pressure and temperature of the flange joint.

b) STEP 2 – Determine an initial flange and clamp geometry (see Figures 4.17.1 and 4.17.2).

c) STEP 3 – Determine the design bolt loads for operating condition, W_o, and the gasket seating and assembly condition, W_g, from paragraph 4.17.4.2.

d) STEP 4 – Determine the flange forces, H, H_p, and H_m from paragraph 4.17.4.2.d, and

$$H_D = 0.785B^2P \tag{4.17.11}$$

$$H_G = \frac{1.571W_o}{\tan\left[\phi + \mu\right]} - \left(H + H_p\right) \tag{4.17.12}$$

$$H_T = H - H_D \tag{4.17.13}$$

e) STEP 5 – Determine the flange moment for the operating condition.

$$M_o = M_D + M_G + M_T + M_F + M_P + M_R \tag{4.17.14}$$

where

$$M_D = H_D \left[\frac{C - (B + g_1)}{2} \right] \tag{4.17.15}$$

$$M_G = H_G h_G \tag{4.17.16}$$

$$M_T = H_T \left[\frac{C}{2} - \frac{(B + G)}{4} \right] \tag{4.17.17}$$

$$M_F = H_D \left(\frac{g_1 - g_0}{2} \right) \tag{4.17.18}$$

$$M_P = PBT\pi \left(\frac{T}{2} - \overline{h} \right) \tag{4.17.19}$$

$$M_R = 1.571 W_o \left(\overline{h} - T + \frac{(C - N)\tan[\phi]}{2} \right) \tag{4.17.20}$$

$$C = \frac{(A + C_i)}{2} \tag{4.17.21}$$

$$\overline{h} = \frac{T^2 g_1 + h_2^2 g_2}{2(T g_1 + h_2 g_2)} \tag{4.17.22}$$

$$h_2 = T - \frac{g_2 \tan[\phi]}{2} \tag{4.17.23}$$

f) STEP 6 – Determine the flange moment for the gasket seating condition

$$M_g = \frac{0.785 W_g (C - G)}{\tan[\phi + \mu]} \tag{4.17.24}$$

g) STEP 7 – Determine the hub factors

$$F_H = 1 + \frac{1.818}{\sqrt{B g_1}} \left[T - \overline{h} + \frac{3.305 I_h}{g_1^2 (0.5B + \overline{g})} \right] \tag{4.17.25}$$

$$I_h = \frac{g_1 T^3}{3} + \frac{g_2 h_2^3}{3} - (g_2 h_2 + g_1 T)\overline{h}^2 \tag{4.17.26}$$

$$\overline{g} = \frac{T g_1^2 + h_2 g_2 (2 g_1 + g_2)}{2(T g_1 + h_2 g_2)} \tag{4.17.27}$$

h) STEP 8 – Determine the reaction shear force at the hub neck for the operating condition.

$$Q_o = \frac{1.818 M_o}{F_H \sqrt{B g_1}} \tag{4.17.28}$$

i) STEP 9 – Determine the reaction shear force at the hub neck for the gasket seating condition.

$$Q_g = \frac{1.818 M_g}{F_H \sqrt{B g_1}} \tag{4.17.29}$$

j) STEP 10 – Determine the clamp factors.

$$e_b = B_c - \frac{C_i}{2} - l_c - X \tag{4.17.30}$$

$$X = \frac{\left(\frac{C_w}{2} - \frac{C_t}{3}\right) C_t^2 - 0.5\left(C_w - C_g\right) l_c^2}{A_c} \tag{4.17.31}$$

$$A_c = A_1 + A_2 + A_3 \tag{4.17.32}$$

$$I_c = \left(\frac{A_1}{3} + \frac{A_2}{4}\right) C_t^2 + \frac{A_3 l_c^2}{3} - A_c X^2 \tag{4.17.33}$$

$$A_1 = \left(C_w - 2C_t\right) C_t \tag{4.17.34}$$

$$A_2 = 1.571 C_t^2 \tag{4.17.35}$$

$$A_3 = \left(C_w - C_g\right) l_c \tag{4.17.36}$$

k) STEP 11 – Determine the hub stress correction factor, f, based on g_1, g_0, h, and B using the equations in Table 4.16.5 and l_m using the following equation.

$$l_m = l_c - 0.5(C - C_i) \tag{4.17.37}$$

l) STEP 12 – Determine the flange and clamp stresses for the operating and gasket seating conditions using the equations in Table 4.17.1.

m) STEP 13 – Check the flange stress acceptance criteria for the operating and gasket seating conditions are shown in Table 4.17.2. If the stress criteria are satisfied, then the design is complete. If the stress criteria are not satisfied, then re-proportion the flange dimensions and go to STEP 2.

4.17.6 Nomenclature

A outside diameter of the hub.

A_b cross-sectional area of the bolts based on the root diameter or the least diameter of the unthreaded portion, if less.

A_c effective clamp cross sectional area.

A_m total minimum required cross-sectional area of the bolts.

A_1 partial clamp area.

A_2 partial clamp area.

A_3 partial clamp area

α hub transition angle, maximum 45°.

B inside diameter of the hub.

B_c radial distance from connection center line to the center of the bolts.

b effective gasket contact width.

b_0 basic gasket seating width.

C diameter of effective clamp-hub reaction circle.

C_i inside diameter of clamp.

C_g effective clamp gap determined at diameter C.

C_t effective clamp thickness subject to the following condition $C_t \geq r$.

C_w clamp width.

e_b radial distance from center of the bolts to the centroid of the clamp cross section

f hub stress correction factor.

F_H factor relating total rotational moment to the reaction moment at the hub neck

g_0 thickness of hub neck at small end.

g_1 thickness of hub neck at intersection with hub shoulder.

g_2 height of hub shoulder.

$\overline{g}$ radial distance from the hub inside diameter B to the hub shoulder ring centroid.

G location of the gasket reaction.

h taper hub length.

h_G radial distance from effective clamp-hub reaction circle to the circle on which H_G acts.

h_2 average thickness of hub shoulder

$\overline{h}$ axial distance from the hub face to the hub shoulder ring centroid

H total hydrostatic end force.

H_D hydrostatic end force on bore area.

H_G difference between total effective axial clamping preload and the sum of total hydrostatic end force and total joint contact surface compression.

H_m total axial gasket seating requirements for makeup.

H_p total joint contact surface compression load.

H_T difference between total hydrostatic end force and hydrostatic end force on bore area.

I_c moment of inertia of clamp relative to neutral axis of entire section.
I_h moment of inertia of hub shoulder relative to its neutral axis
L_a distance from *W* to the point where the clamp lug joins the clamp body.
L_h clamp lug height.
L_w clamp lug width.
l_c effective clamp lip length.
l_m effective clamp lip moment arm.
M_D moment due to H_D.
M_F offset moment.
M_G moment due to H_G.
M_P pressure moment.
M_R radial clamp equilibriating moment.
M_T moment due to H_T.
M_o flange design moment for the operating condition.
M_g flange design moment for the gasket seating condition.
μ friction angle.
m gasket factor.
N outside diameter of hub neck.
m gasket factor.
P design pressure.
ϕ clamp shoulder angle, maximum 40°.
Q_g reaction shear force at the hub neck for the gasket seating condition.
Q_o reaction shear force at the hub neck for the design operating condition.
r clamp or hub cross section corner radius.
S_{bg} allowable stress from Annex 3.A for the bolt evaluated at the gasket seating temperature.
S_{bo} allowable stress from Annex 3.A 3 for the bolt evaluated at the design temperature.
S_{cg} allowable stress from Annex 3.A for the clamp evaluated at the gasket seating temperature.
S_{co} allowable stress from Annex 3.A for the clamp evaluated at the design temperature.
S_{hg} allowable stress from Annex 3.A for the hub evaluated at the gasket seating temperature.
S_{ho} allowable stress from Annex 3.A for the hub evaluated at the design temperature.
S_{1g} hub longitudinal stress on outside at hub neck for the design gasket seating condition.
S_{1o} hub longitudinal stress on outside at hub neck for the design operating condition.
S_{2g} maximum Lame hoop stress at bore of hub for the design gasket seating condition.
S_{2o} maximum Lame hoop stress at bore of hub for the design operating condition.
S_{3g} maximum hub shear stress at shoulder for the design gasket seating condition.
S_{3o} maximum hub shear stress at shoulder for the design operating condition.
S_{4g} maximum radial hub shear stress in neck for the design gasket seating condition.

S_{4o} maximum radial hub shear stress in neck for the design operating condition.

S_{5g} clamp longitudinal stress at clamp body inner diameter for the design gasket seatingcondition.

S_{5o} clamp longitudinal stress at clamp body inner diameter for the design operating condition.

S_{6g} clamp tangential stress at clamp body outer diameter for the design gasket seating condition.

S_{6o} clamp tangential stress at clamp body outer diameter for the design operating condition.

S_{7g} maximum shear stress in clamp lips for the design gasket seating condition.

S_{7o} maximum shear stress in clamp lips for the design operating condition.

S_{8g} clamp lug bending stress for the design gasket seating condition.

S_{8o} clamp lug bending stress for the design operating condition.

S_{9g} effective bearing stress between clamp and hub for the design gasket seating condition.

S_{9o} effective bearing stress between clamp and hub for the design operating condition.

T thickness of hub shoulder.

W_g clamp connection design bolt load for the gasket seating and assembly condition.

W_{g1} clamp connection design bolt load for the gasket seating condition.

W_{g2} clamp connection design bolt load for the assembly condition.

W_o clamp connection design bolt load for the design condition.

X clamp dimension to neutral axis.

y gasket seating stress.

Z clamp-hub taper angle.

4.17.7 Tables

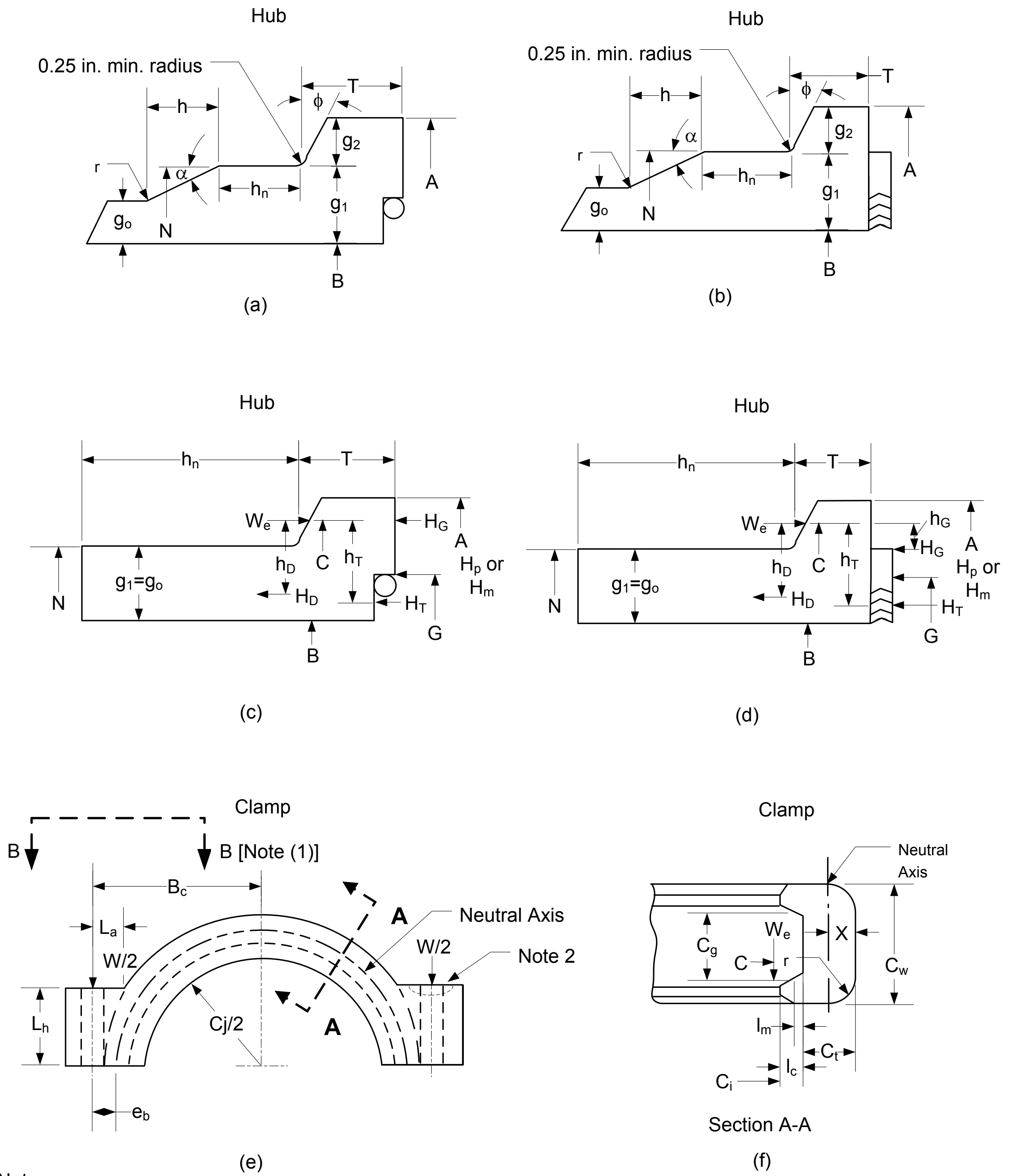
Table 4.17.1 – Flange Stress Equations

Location	Stress Equations	
	Operating Condition	**Gasket Seating/Assembly Conditions**
Flange	$S_{1o} = f\left[\frac{PB^2}{4g_1\left(B+g_1\right)} + \frac{1.91M_o}{g_1^2\left(B+g_1\right)F_H}\right]$ $S_{2o} = P\left(\frac{N^2+B^2}{N^2-B^2}\right)$ $S_{3o} = \frac{0.75W_o}{T\left(B+2g_1\right)\tan\left(\phi-\mu\right)}$ $S_{4o} = \frac{0.477Q_o}{g_1\left(B+2g_1\right)}$	$S_{1g} = f\left[\frac{1.91M_g}{g_1^2\left(B+g_1\right)F_H}\right]$ $S_{2g} = 0.0$ $S_{3g} = \frac{0.75W_g}{T\left(B+2g_1\right)\tan\left(\phi+\mu\right)}$ $S_{4g} = \frac{0.477Q_g}{g_1\left(B+2g_1\right)}$
Clamp	$S_{5o} = \frac{W_o}{2C\tan\left(\phi-\mu\right)}\left[\frac{1}{C_t} + \frac{3\left(C_t+2l_m\right)}{C_t^2}\right]$ $S_{6o} = \frac{W_o}{2}\left[\frac{1}{A_c} + \frac{\lvert e_b\rvert\cdot\left(C_t-X\right)}{I_c}\right]$ $S_{7o} = \frac{1.5W_o}{\left(C_w-C_g\right)C\tan\left(\phi-\mu\right)}$ $S_{8o} = \frac{3W_oL_a}{L_wL_h^2}$ $S_{9o} = \frac{W_o}{\left(A-C_i\right)C\tan\left(\phi-\mu\right)}$	$S_{5g} = \frac{W_g}{2C\tan\left(\phi+\mu\right)}\left[\frac{1}{C_t} + \frac{3\left(C_t+2l_m\right)}{C_t^2}\right]$ $S_{6g} = \frac{W_g}{2}\left[\frac{1}{A_c} + \frac{\lvert e_b\rvert\cdot\left(C_t-X\right)}{I_c}\right]$ $S_{7g} = \frac{1.5W_g}{\left(C_w-C_g\right)C\tan\left(\phi+\mu\right)}$ $S_{8g} = \frac{3W_gL_a}{L_wL_h^2}$ $S_{9g} = \frac{W_g}{\left(A-C_i\right)C\tan\left(\phi+\mu\right)}$

Table 4.17.2 – Flange Stress Acceptance Criteria

Location	Stress Acceptance Criteria	
	Operating Condition	**Gasket Seating/Assembly Conditions**
Flange	$S_{1o} \le 1.5 S_{ho}$	$S_{1g} \le 1.5 S_{hg}$
	$S_{2o} \le S_{ho}$	$S_{2g} \le S_{hg}$
	$S_{3o} \le 0.8 S_{ho}$	$S_{3g} \le 0.8 S_{hg}$
	$S_{4o} \le 0.8 S_{ho}$	$S_{4g} \le 0.8 S_{hg}$
Clamp	$S_{5o} \le 1.5 S_{co}$	$S_{5g} \le 1.5 S_{cg}$
	$S_{6o} \le 1.5 S_{co}$	$S_{6g} \le 1.5 S_{cg}$
	$S_{7o} \le 0.8 S_{co}$	$S_{7g} \le 0.8 S_{cg}$
	$S_{8o} \le S_{co}$	$S_{8g} \le S_{cg}$
	$S_{9o} \le 1.6 \min\left[S_{ho},\ S_{co}\right]$	$S_{9g} \le 1.6 \min\left[S_{hg},\ S_{cg}\right]$

4.17.8 Figures



Notes:
1) See Figure 4.17.2 for section B-B
2) Clamp may have spherical depressions at bolt holes to facilitate the use of spherical nuts

Figure 4.17.1 – Typical Hub and Clamp Configuration

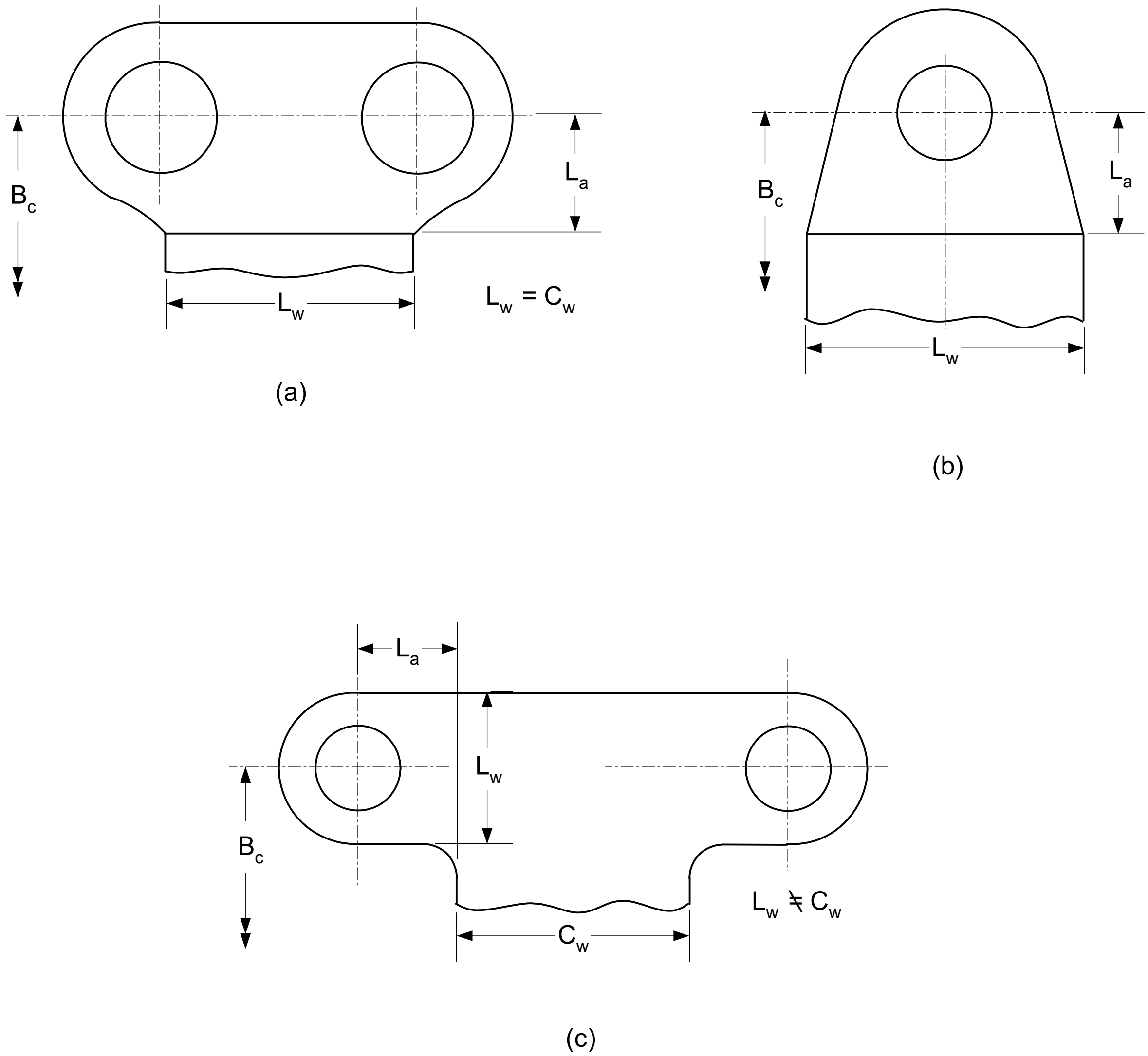


Figure 4.17.2 – Typical Clamp Lugs Configurations

4.18 Design Rules for Shell and Tube Heat Exchangers

4.18.1 Scope

The rules in paragraph 4.18 cover the minimum requirements for design, fabrication and inspection of shell-and-tube heat exchangers.

4.18.2 Terminology

a) U-Tube Heat Exchanger – A heat exchanger with one stationary tubesheet attached to the shell and channel. The heat exchanger contains a bundle of U-tubes attached to the tubesheet (see Figure 4.18.1.Sketch (a)).

b) Fixed Tubesheet Heat Exchanger – A heat exchanger with two stationary tubesheets, each attached to the shell and channel. The heat exchanger contains a bundle of straight tubes connecting both tubesheets (see Figure 4.18.1. Sketch (b)).

c) Floating Tubesheet Heat Exchanger – A heat exchanger with one stationary tubesheet attached to the shell and channel, and one floating tubesheet that can move axially. The heat exchanger contains a bundle of straight tubes connecting both tubesheets (see Figure 4.18.1. Sketch (c)).

4.18.3 General Design Considerations

a) The design of all components shall be in accordance with the applicable rules of all Parts of this Division.

b) The design of a bolted flat cover where the cover bears against a gasket at the pass partition shall consider the effects of deflection.

c) The design of flanges shall consider the effects of pass partition gasketing in determining the minimum required bolt loads, W_o and W_g, of paragraph 4.16. When the tubesheet is gasketed between the shell and channel flanges, the shell and channel flange bolt loads are identical and shall be treated as flange pairs in accordance with paragraph 4.16.

d) Rules for U-tube heat exchangers are covered in paragraph 4.18.7.

e) Rules for fixed tubesheet heat exchangers are covered in paragraph 4.18.8.

f) Rules for floating tubesheet heat exchangers are covered in paragraph 4.18.9.

g) Distribution and vapor belts where the shell is not continuous across the belt shall be designed in accordance with paragraph 4.18.12.

h) Requirements for tubes shall be as follows.

   1) The allowable axial tube stresses in fixed and floating tubesheet heat exchangers shall be in accordance with paragraphs 4.18.8 and 4.18.9.

   2) The thickness of U-tubes after forming shall not be less than the design thickness.

4.18.4 General Conditions of Applicability for Tubesheets

a) The tubesheet shall be flat and circular. It shall be of uniform thickness, except that the flanged extension may differ in thickness as determined in paragraph 4.18.5. The tubesheet shall be uniformly perforated over a nominally circular area, in either equilateral triangular or square patterns. However, untubed lanes for pass partitions are permitted

b) The channel component integral with the tubesheet (Configurations a, e, f and A for U-Tube, Fixed, and Floating Tubesheets) shall be either a cylinder or a hemispherical head (see Figure 4.18.15). The hemispherical head rules shall be used when the head is attached directly to the tubesheet and there are no cylindrical straight sections between the head and the tubesheet.

c) The tube side and shell side pressures are assumed to be uniform. These rules do not cover weight loadings or pressure drop.

d) If these conditions of applicability are not satisfied, the design shall be in accordance with Part 5.

4.18.5 Tubesheet Flanged Extension

4.18.5.1 Scope

The following rules cover the determination of the required thickness of the tubesheet flanged extension when bolt loads are transmitted to the extended edge of the tubesheet. The required thickness of the extended portion of the tubesheet may differ from that required for the interior of the tubesheet as calculated in paragraphs 4.18.7, 4.18.8, and 4.18.9.

4.18.5.2 Conditions of Applicability

The general conditions of applicability given in paragraph 4.18.4 apply.

4.18.5.3 Design Considerations

a) The designer should take appropriate consideration of the stresses resulting from the pressure test required in paragraph 4.1.6.2 and Part 8. Special consideration shall be required for tubesheets that are gasketed on both sides when the pressure test in each chamber is conducted independently, and the bolt loading is only applied to the flanged extension during the pressure test.

b) If the tubesheet is grooved for a peripheral gasket, the thinnest section of the flanged extension shall not be less than h_r. Figure 4.18.16 depicts h_r for some representative configurations

c) The thickness h_r shall be calculated for both operating and gasket seating conditions, and the greater of the two values shall be used. For operating conditions, the values of S at the design temperature and $W = W_o$ (see paragraph 4.16) shall be used. For gasket seating, the values of S at atmospheric temperature and $W = W_g$ (see paragraph 4.16) shall be used.

4.18.5.4 Calculation Procedure

The procedure for calculating the minimum required thickness of the tubesheet flanged extension h_r is as follows:

$$h_r = \left(\frac{1.9 W h_g}{S \cdot G} \right)^{0.5} \tag{4.18.1}$$

4.18.6 Tubesheet Characteristics

4.18.6.1 Scope

These rules cover the determination of the ligament efficiencies, effective depth of the tube side pass partition groove, and effective elastic constants to be used in the calculation of U-tube, fixed, and floating tubesheets.

4.18.6.2 Conditions of Applicability

The general conditions of applicability given in paragraph 4.18.4 apply.

4.18.6.3 Design Considerations

a) Elastic moduli and allowable stresses shall be taken at the design temperatures. However, for cases involving thermal loading, it is permitted to use the operating temperatures instead of the design temperatures.

b) When the values calculated in this section are to be used for fixed tubesheets, they shall be determined in both the corroded and uncorroded conditions.

c) The tube expansion depth ratio given by Equation (4.18.4) may be either calculated or chosen as a constant.

4.18.6.4 Calculation Procedure

a) Determination of Effective Dimensions and Ligament Efficiencies – From the geometry (see Figures 4.18.2 and 4.18.3) and material properties of the exchanger, calculate the required parameters in accordance with paragraphs 4.18.6.4.a.1 or 4.18.6.4.a.2.

1) For geometries where the tubes extend through the tubesheet (see Figure 4.18.2.Sketch (b)), calculate the following parameters.

$$D_o = 2r_o + d_t \tag{4.18.2}$$

$$\mu = \frac{p - d_t}{p} \tag{4.18.3}$$

$$\rho = \frac{l_{tx}}{h} \qquad where \;\; 0 \le \rho \le 1 \tag{4.18.4}$$

$$d^* = \max\left[\left\{d_t - 2t_t\left(\frac{E_{tT}}{E}\right)\left(\frac{S_{tT}}{S}\right)\rho\right\},\ \left(d_t - 2t_t\right)\right] \tag{4.18.5}$$

$$p^* = p\left(1 - \frac{4\cdot\min\left[A_L,\left(4D_o p\right)\right]}{\pi D_o^{\,2}}\right)^{-0.5} \tag{4.18.6}$$

$$\mu^* = \frac{p^* - d^*}{p^*} \tag{4.18.7}$$

$$h_g' = \max\left[\left(h_g - c_t\right)\ ,\ 0.0\right] \tag{4.18.8}$$

2) For tubes welded to the backside of the tubesheet (see Figure 4.18.2. Sketch (d)), calculate the following parameters.

$$D_o = 2r_o + d \tag{4.18.9}$$

$$\mu = \frac{p - d}{p} \tag{4.18.10}$$

$$p^* = p\left(1 - \frac{4\cdot\min\left[A_L, \left(4D_0 p\right)\right]}{\pi D_o^{\,2}}\right)^{-0.5} \tag{4.18.11}$$

$$\mu^* = \frac{p^* - d}{p^*} \tag{4.18.12}$$

$$h_g^{'} = \max\left[\left(h_g - c_t\right),\, 0.0\right] \tag{4.18.13}$$

b) Determination of Effective Elastic Properties – Determine the values for E^*/E and ν^* based on μ^* and h/p using Table 4.18.1 (equilateral triangular pattern) or Table 4.18.2 (square pattern).

4.18.7 Rules for the Design of U-Tube Tubesheets

4.18.7.1 Scope

These rules cover the design of tubesheets for U-tube heat exchangers. The tubesheet may have one of the six configurations shown in Figure 4.18.4.

a) Configuration a – tubesheet integral with shell and channel.

b) Configuration b – tubesheet integral with shell and gasketed with channel, extended as a flange.

c) Configuration c – tubesheet integral with shell and gasketed with channel, not extended as a flange.

d) Configuration d – tubesheet gasketed with shell and channel, extended or not extended as a flange.

e) Configuration e – tubesheet gasketed with shell and integral with channel, extended as a flange.

f) Configuration f – tubesheet gasketed with shell and integral with channel, not extended as a flange.

4.18.7.2 Conditions of Applicability

The general conditions of applicability given in paragraph 4.18.4 apply.

4.18.7.3 Design Considerations

a) The various loading conditions to be considered shall include the normal operating conditions, the startup conditions, the shutdown conditions, and the upset conditions, which may govern the design of the tubesheet.

   1) For each of these conditions, the following loading cases shall be considered:

      i) Loading Case 1 – Tube side pressure P_t acting only $\left(P_s = 0\right)$.

      ii) Loading Case 2 – Shell side pressure P_s acting only $\left(P_t = 0\right)$.

      iii) Loading Case 3 – Tube side pressure P_t and shell side pressure P_s acting simultaneously.

   2) When vacuum exists, each loading case shall be considered with and without the vacuum.

   3) When differential design pressure is specified by the user, the design shall be based only on Loading Case 3. If the tube side is the higher-pressure side, P_t shall be the tube side design pressure and P_s shall be P_t less the differential design pressure. If the shell side is the higher-pressure side, P_s shall be the shell side design pressure and P_t shall be P_s less the differential

design pressure.

4) The designer should take appropriate consideration of the stresses resulting from the pressure test required by paragraph 4.1.6.2 and Part 8.

b) As the calculation procedure is iterative, a value h shall be assumed for the tubesheet thickness to calculate and check that the maximum stresses in tubesheet, shell, and channel are within the maximum permissible stress limits.

c) The designer shall consider the effect of deflections in the tubesheet design, especially when the tubesheet thickness h is less than the tube diameter.

d) The designer may consider the tubesheet as simply supported in accordance with paragraph 4.18.7.5

4.18.7.4 Calculation Procedure

a) STEP 1 – Determine D_o, μ, μ^*, and h_g' from paragraph 4.18.6.4.a.

b) STEP 2 – Calculate the diameter ratios ρ_s and ρ_c using the following equations.

$$\rho_s = \frac{D_s}{D_o} \qquad Configurations - a,b,c \tag{4.18.14}$$

$$\rho_s = \frac{G_s}{D_o} \qquad Configurations - d,e,f \tag{4.18.15}$$

$$\rho_c = \frac{D_c}{D_o} \qquad Configurations - a,e,f \tag{4.18.16}$$

$$\rho_c = \frac{G_c}{D_o} \qquad Configurations - b,c,d \tag{4.18.17}$$

For each loading case, calculate moment M_{TS} due to pressures P_s and P_t acting on the unperforated tubesheet rim.

$$M_{TS} = \frac{D_o^2\left[\left(\rho_s - 1\right)\left(\rho_s^2 + 1\right)P_s - \left(\rho_c - 1\right)\left(\rho_c^2 + 1\right)P_t\right]}{16} \tag{4.18.18}$$

c) STEP 3 – Calculate h/p. Determine E^*/E and ν^* using paragraph 4.18.6.4.b. For Configurations a, b, c, e and f, proceed to STEP 4. For Configuration d, proceed to STEP 5.

d) STEP 4 – Calculate the shell coefficients.

1) Configurations a, b and c:

$$\beta_s = \frac{\left[12\left(1-\nu_s^2\right)\right]^{0.25}}{\left[\left(D_s + t_s\right)t_s\right]^{0.5}} \tag{4.18.19}$$

$$k_s = \frac{\beta_s E_s t_s^3}{6\left(1-\nu_s^2\right)} \tag{4.18.20}$$

$$\lambda_s = \frac{6\ D_s k_s}{h^3}\left(1+h\beta_s+\frac{h^2\beta_s^2}{2}\right) \tag{4.18.21}$$

$$\delta_s = \frac{D_s^2}{4E_s\ t_s}\left(1-\frac{\nu_s}{2}\right) \tag{4.18.22}$$

$$\omega_s = \rho_s k_s \beta_s \delta_s\left(1+h\beta_s\right) \tag{4.18.23}$$

2) Configurations a, e and f:

$$\beta_c = \frac{\left[12\left(1-\nu_c^2\right)\right]^{0.25}}{\left[\left(D_c+t_c\right)t_c\right]^{0.5}} \tag{4.18.24}$$

$$k_c = \frac{\beta_c E_c t_c^3}{6\left(1-\nu_c^2\right)} \tag{4.18.25}$$

$$\lambda_c = \frac{6\ D_c k_c}{h^3}\left(1+h\beta_c+\frac{h^2\beta_c^2}{2}\right) \tag{4.18.26}$$

$$\delta_c = \frac{D_c^2}{4E_c\ t_c}\left(1-\frac{\nu_c}{2}\right) \qquad \textit{For a Cylinder} \tag{4.18.27}$$

$$\delta_c = \frac{D_c^2}{4E_c\ t_c}\left(\frac{1-\nu_c}{2}\right) \qquad \textit{For a Hemispherical Head} \tag{4.18.28}$$

$$\omega_c = \rho_c k_c \beta_c \delta_c\left(1+h\ \beta_c\right) \tag{4.18.29}$$

e) STEP 5 – Calculate the diameter ratio, K, and the coefficient F.

$$K = \frac{A}{D_o} \tag{4.18.30}$$

$$F = \frac{\left(1-\nu^*\right)\left(\lambda_s+\lambda_c+E\ln K\right)}{E^*} \qquad Configuration-a \tag{4.18.31}$$

$$F = \frac{\left(1-\nu^*\right)\left(\lambda_s+E\ln K\right)}{E^*} \qquad Configurations-b,c \tag{4.18.32}$$

$$F = \frac{(1-\nu^*)(E \ln K)}{E^*} \qquad Configuration-d \qquad (4.18.33)$$

$$F = \frac{(1-\nu^*)(\lambda_c + E \ln K)}{E^*} \qquad Configurations-e, f \qquad (4.18.34)$$

f) STEP 6 – For each loading case, calculate the moment M^* acting on the unperforated tubesheet rim.

$$M^* = M_{TS} + \omega_c P_t - \omega_s P_s \qquad Configuration-a \qquad (4.18.35)$$

$$M^* = M_{TS} - \omega_s P_s - \frac{(C-G_c)W_c}{2\pi D_o} \qquad Configuration-b \qquad (4.18.36)$$

$$M^* = M_{TS} - \omega_s P_s - \frac{(G_1-G_c)W_c}{2\pi D_o} \qquad Configuration-c \qquad (4.18.37)$$

$$M^* = M_{TS} + \frac{(G_c-G_s)W_{\max}}{2\pi D_o} \qquad Configuration-d \qquad (4.18.38)$$

$$W_{\max} = \max\left[W_s, W_c\right] \qquad (4.18.39)$$

$$M^* = M_{TS} + \omega_c P_t + \frac{(C-G_s)W_s}{2\pi D_o} \qquad Configuration-e \qquad (4.18.40)$$

$$M^* = M_{TS} + \omega_c P_t + \frac{(G_1-G_s)W_s}{2\pi D_o} \qquad Configuration-f \qquad (4.18.41)$$

g) STEP 7 – For each loading case, calculate the maximum bending moments acting on the tubesheet at the periphery, M_p, and at the center, M_o.

$$M_p = \frac{M^* - \dfrac{D_o^2 F(P_s - P_t)}{32}}{1+F} \qquad (4.18.42)$$

$$M_o = M_p + \frac{D_o^2(3+\nu^*)(P_s - P_t)}{64} \qquad (4.18.43)$$

Determine the maximum bending moment M acting on the tubesheet.

$$M = \max\left[\left|M_o\right|, \left|M_p\right|\right] \qquad (4.18.44)$$

h) STEP 8 – For each loading case, check the tubesheet bending stress.

1) Calculate the bending stress

$$\sigma = \frac{6M}{\mu^* \left(h - h_g'\right)^2} \tag{4.18.45}$$

2) Acceptance criteria

If $\sigma \le 2S$, the assumed tubesheet thickness is acceptable for bending. Otherwise, increase the assumed tubesheet thickness h and return to STEP 1.

i) STEP 9 – For each loading case, check the average shear stress in the tubesheet at the outer edge of the perforated region.

1) Calculate the average shear stress. The shear stress may be conservatively calculated using the following equation.

$$\tau = \left(\frac{1}{4\mu}\right)\left(\frac{D_o}{h}\right) \cdot \left|P_s - P_t\right| \tag{4.18.46}$$

If $\left|P_s - P_t\right| > \dfrac{3.2S\mu h}{D_o}$, the shear stress may be more accurately calculated by the following equation.

$$\tau = \left(\frac{1}{4\mu}\right)\left(\frac{1}{h}\left\{\frac{4A_p}{C_p}\right\}\right) \cdot \left|P_s - P_t\right| \tag{4.18.47}$$

2) Acceptance criteria

If $\tau \le 0.8S$, the assumed tubesheet thickness is acceptable for shear. Otherwise, increase the assumed tubesheet thickness h and return to STEP 1.

3) For Configurations a, b, c, e, and f, proceed to STEP 10. For Configuration d, the calculation procedure is complete.

j) STEP 10 – For each loading case, check the stresses in the shell and/or channel integral with the tubesheet. .

1) Configurations a, b and c – The shell shall have a uniform thickness of t_s for a minimum length of $1.8\sqrt{D_s\, t_s}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{s,m}$, the bending stress, $\sigma_{s,b}$,and total axial stress, σ_s, in the shell at its junction to the tubesheet.

$$\sigma_{s,m} = \frac{D_s^2 P_s}{4t_s\left(D_s + t_s\right)} \tag{4.18.48}$$

$$\sigma_{s,b} = \frac{6k_s}{t_s^2}\left[\beta_s\delta_s P_s + \frac{6\left(1-\nu^*\right)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1+\frac{h\beta_s}{2}\right)\left(M_p + \frac{D_o^2}{32}\left(P_s - P_t\right)\right)\right] \tag{4.18.49}$$

$$\sigma_s = \left|\sigma_{s,m}\right| + \left|\sigma_{s,b}\right| \tag{4.18.50}$$

2) Configurations a, e and f – A cylindrical channel shall have a uniform thickness of t_c for a minimum length of $1.8\sqrt{D_c\ t_c}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{c,m}$, the bending stress, $\sigma_{c,b}$, and total axial stress, σ_c, in the channel at its junction to the tubesheet.

$$\sigma_{c,m} = \frac{D_c^2 P_t}{4t_c\left(D_c + t_c\right)} \tag{4.18.51}$$

$$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c \delta_c P_t - \frac{6\left(1-\nu^*\right)}{E^*}\left(\frac{D_o}{h^3}\right)\left(1+\frac{h\beta_c}{2}\right)\left(M_p + \frac{D_o^2}{32}\left(P_s - P_t\right)\right)\right] \tag{4.18.52}$$

$$\sigma_c = \left|\sigma_{c,m}\right| + \left|\sigma_{c,b}\right| \tag{4.18.53}$$

3) Acceptance Criteria

i) Configuration a – If $\sigma_s \le 1.5S_s$ and $\sigma_c \le 1.5S_c$, the shell and channel designs are acceptable and the calculation procedure is complete. Otherwise, proceed to STEP 11.

ii) Configurations b and c – If $\sigma_s \le 1.5S_s$, the shell design is acceptable and the calculation procedure is complete. Otherwise, proceed to STEP 11.

iii) Configurations e and f – If $\sigma_c \le 1.5S_c$, the channel design is acceptable and the calculation procedure is complete. Otherwise, proceed to STEP 11.

k) STEP 11 – The design shall be reconsidered. One or a combination of the following three options may be used.

1) Option 1 – Increase the assumed tubesheet thickness h and return to STEP 1.

2) Option 2 – Increase the integral shell and/or channel thickness as follows, and return to STEP 2.

i) Configurations a, b and c – If $\sigma_s > 1.5S_s$, increase the shell thickness t_s.

ii) Configurations a, e and f – If $\sigma_c > 1.5S_c$, increase the channel thickness t_c.

3) Option 3 –Perform a simplified elastic-plastic calculation for each applicable loading case by using a reduced effective modulus for the integral shell and/or channel to reflect the anticipated load shift resulting from plastic action at the integral shell and/or channel-to-tubesheet junction. This may result in a higher tubesheet bending stress, σ. This option shall not be used at temperatures where the time-dependent properties govern the allowable stress.

i) Configuration a – This option may only be used when $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$. In STEP 4, if $\sigma_s > 1.5S_s$, replace E_s with $E_s^* = E_s\sqrt{1.5S_s/\sigma_s}$ and recalculate k_s and λ_s. If $\sigma_c > 1.5S_c$, replace E_c with $E_c^* = E_c\sqrt{1.5S_c/\sigma_c}$ and recalculate k_c and λ_c.

ii) Configurations b and c – This option may only be used when $\sigma_s \le S_{PS,s}$. In STEP 4, replace E_s with $E_s^* = E_s\sqrt{1.5S_s/\sigma_s}$ and recalculate k_s and λ_s.

iii) Configurations e and f – This option may only be used when $\sigma_c \le S_{PS,c}$. In STEP 4, replace E_c with $E_c^* = E_c\sqrt{1.5S_c/\sigma_c}$ and recalculate k_c and λ_c.

After making the above changes, perform STEPs 5 and 7, and recalculate the tubesheet bending stress σ given in STEP 8. If $\sigma \le 2S$, the assumed tubesheet thickness h is acceptable and the design is complete. Otherwise, the design shall be reconsidered by using Option 1 or 2.

4.18.7.5 Calculation Procedure for Simply Supported U-Tube Tubesheets

4.18.7.5.1 Scope

This procedure describes how to use the rules of paragraph 4.18.7.4 when the effect of the stiffness of the integral channel and/or shell is not considered.

4.18.7.5.2 Conditions of Applicabiltiy

This calculation procedure applies only when the tubeheet is integral with the shell or channel (Configurations a,b,c,e,and f).

4.18.7.5.3 Calculation Procedure

The calculation procedure outlined in 4.18.7.4 shall be performed accounting for the following modifications.

a) Perform STEPs 1 through 9.

b) Perform STEP 10 except as follows:

1) The shell (Configuration a,b,and c) is not required to meet a minimum length requirement.

2) The channel (Configurations a,e,and f) is not required to meet a minimum length requirements.

3) Acceptance Criteria

i) Configuration a: If $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$, then the shell and channel are acceptable. Otherwise increase the thickness of the overstressed component(s) (shell and/or channel) and return to STEP 1.

ii) Configuration b and c : If $\sigma_s \le S_{PS,s}$ then the shell is acceptable. Otherwise increase the thickness of the shell and return to STEP 1.

iii) Configuration e and f : If $\sigma_c \le S_{PS,c}$ then the channel is acceptable. Otherwise increase the thickness of the channel and return to STEP 1.

c) Do not perform STEP 11

d) Repeat STEPs 1 - 8 with the following changes until the tubesheet criteria have been met:

1) STEP 4

i) Configurations a, b, and c: $\beta_s = 0,\ k_s = 0,\ \lambda_s = 0,\ \delta_s = 0$.

ii) Configurations a, e, and f: $\beta_c = 0,\ k_c = 0,\ \lambda_c = 0,\ \delta_c = 0$.

2) STEP 7 $M = |M_o|$

4.18.8 Rules for the Design of Fixed Tubesheets

4.18.8.1 Scope

These rules cover the design of tubesheets for fixed tubesheet heat exchangers. The tubesheets may have one of the four configurations shown in Figure 4.18.5.

a) Configuration a – tubesheet integral with shell and channel.

b) Configuration b – tubesheet integral with shell and gasketed with channel, extended as a flange.

c) Configuration c – tubesheet integral with shell and gasketed with channel, not extended as a flange.

d) Configuration d – tubesheet gasketed with shell and channel, extended or not extended as a flange.

4.18.8.2 Conditions of Applicability

The two tubesheets shall have the same thickness, material and edge conditions.

4.18.8.3 Design Considerations

a) It is generally not possible to determine, by observation, the most severe condition of coincident pressure, temperature and differential thermal expansion. Thus, it is necessary to evaluate all the anticipated loading conditions to ensure that the worst load combination has been considered in the design. The various loading conditions to be considered shall include the normal operating conditions, the startup conditions, the shutdown conditions, and the upset conditions, which may govern the design of the main components of the heat exchanger (i.e., tubesheets, tubes, shell, channel, tube-to-tubesheet joint).

1) For each of these conditions, the following loading cases shall be considered to determine the effective pressure P_e to be used in the design equations:

i) Loading Case 1 – Tube side pressure P_t acting only $(P_s = 0)$ without differential thermal expansion.

ii) Loading Case 2 – Shell side pressure P_s acting only $(P_t = 0)$ without differential thermal expansion.

iii) Loading Case 3 – Tube side pressure P_t and shell side pressure P_s acting simultaneously, without differential thermal expansion.

iv) Loading Case 4 – Differential thermal expansion acting only $(P_s = 0)$ and $(P_t = 0)$.

v) Loading Case 5 – Tube side pressure P_t acting only $(P_s = 0)$ with differential thermal expansion.

vi) Loading Case 6 – Shell side pressure P_s acting only $(P_t = 0)$ with differential thermal expansion.

vii) Loading Case 7 – Tube side pressure P_t and shell side pressure P_s acting simultaneously, with differential thermal expansion.

2) When vacuum exists, each loading case shall be considered with and without the vacuum.

3) When differential design pressure is specified by the user, the design shall be based only on Loading Cases 3, 4, and 7. If the tube side is the higher-pressure side, P_t shall be the tube side

design pressure and P_s shall be P_t less the differential design pressure. If the shell side is the higher-pressure side, P_s shall be the shell side design pressure and P_t shall be P_s less the differential design pressure.

4) The designer should take appropriate consideration of the stresses resulting from the pressure test required by paragraph 4.1.6.2 and Part 8.

b) Elastic moduli, yield strengths, and allowable stresses shall be taken at design temperatures. However for cases involving thermal loading (Loading Cases 4, 5, 6, and 7), it is permitted to use the operating temperatures instead of the design temperatures.

c) As the calculation procedure is iterative, a value h shall be assumed for the tubesheet thickness to calculate and check that the maximum stresses in tubesheet, tubes, shell, and channel are within the maximum permissible stress limits and that the resulting tube-to-tubesheet joint load is acceptable.

d) Because any increase of tubesheet thickness may lead to overstresses in the tubes, shell, channel, or tube-to-tubesheet joint, a final check shall be performed, using in the equations the nominal thickness of tubesheet, tubes, shell, and channel, in both corroded and uncorroded conditions.

e) The designer shall consider the following:

1) The effect of deflections in the tubesheet design, especially when the tubesheet thickness h is less than the tube diameter.

2) The effect of radial differential thermal expansion between the tubesheet and integral shell or channel (Configurations a, b, and c) in accordance with paragraph 4.18.8.7.

f) The designer may consider the tubesheet as simply supported in accordance with 4.18.8.8.

4.18.8.4 Calculation Procedure

a) STEP 1 – Determine D_o, μ, μ^*, and $h_g^{'}$ from paragraph 4.18.6.4.a. For Loading Cases 4, 5, 6, and 7, $h_g^{'} = 0$. Calculate the following quantities.

$$a_o = \frac{D_o}{2} \tag{4.18.54}$$

$$a_c = \frac{D_c}{2} \qquad Configuration - a \tag{4.18.55}$$

$$a_c = \frac{G_c}{2} \qquad Configurations - b, c, d \tag{4.18.56}$$

$$a_s = \frac{D_s}{2} \qquad Configurations - a, b, c \tag{4.18.57}$$

$$a_s = \frac{G_s}{2} \qquad Configuration - d \tag{4.18.58}$$

$$\rho_s = \frac{a_s}{a_o} \tag{4.18.59}$$

$$\rho_c = \frac{a_c}{a_o} \tag{4.18.60}$$

$$x_s = 1 - N_t \left(\frac{d_t}{2\ a_o} \right)^2 \tag{4.18.61}$$

$$x_t = 1 - N_t \left(\frac{d_t - 2t_t}{2\ a_o} \right)^2 \tag{4.18.62}$$

b) STEP 2 – Calculate the following parameters.

1) The shell axial stiffness and the tube axial stiffness.

$$K_s = \frac{\pi\ t_s \left(D_s + t_s\right) E_s}{L} \tag{4.18.63}$$

$$K_t = \frac{\pi\ t_t \left(d_t - t_t\right) E_t}{L} \tag{4.18.64}$$

2) The stiffness factors.

$$K_{s,t} = \frac{K_s}{N_t K_t} \tag{4.18.65}$$

$$J = \frac{K_J}{K_J + K_S} \tag{4.18.66}$$

3) The shell coefficients for Configurations a, b and c.

$$\beta_s = \frac{\left[12\left(1 - \nu_s^2\right)\right]^{0.25}}{\left[\left(D_s + t_s\right) t_s\right]^{0.5}} \tag{4.18.67}$$

$$k_s = \frac{\beta_s E_s t_s^3}{6\left(1 - \nu_s^2\right)} \tag{4.18.68}$$

$$\lambda_s = \frac{6\ D_s k_s}{h^3} \left(1 + h\beta_s + \frac{h^2 \beta_s^2}{2}\right) \tag{4.18.69}$$

$$\delta_s = \frac{D_s^2}{4E_s\ t_s} \left(1 - \frac{\nu_s}{2}\right) \tag{4.18.70}$$

For Configuration d, $\beta_s = k_s = \lambda_s = \delta_s = 0$.

4) The channel coefficients for Configuration a.

$$\beta_c = \frac{\left[12\left(1-\nu_c^2\right)\right]^{0.25}}{\left[\left(D_c+t_c\right)t_c\right]^{0.5}} \tag{4.18.71}$$

$$k_c = \frac{\beta_c E_c t_c^3}{6\left(1-\nu_c^2\right)} \tag{4.18.72}$$

$$\lambda_c = \frac{6\,D_c k_c}{h^3}\left(1+h\beta_c+\frac{h^2\beta_c^2}{2}\right) \tag{4.18.73}$$

$$\delta_c = \frac{D_c^2}{4E_c\ t_c}\left(1-\frac{\nu_c}{2}\right) \qquad \textit{For a Cylinder} \tag{4.18.74}$$

$$\delta_c = \frac{D_c^2}{4E_c\ t_c}\left(\frac{1-\nu_c}{2}\right) \qquad \textit{For a Hemispherical Head} \tag{4.18.75}$$

For Configurations b, c, and d, $\beta_c = k_c = \lambda_c = \delta_c = 0$.

c) STEP 3 – Calculate h/p. Determine E^*/E and ν^* using paragraph 4.18.6.4.b. Calculate X_a.

$$X_a = \left[\frac{24\left(1-\left(\nu^*\right)^2\right)N_t E_t t_t\left(d_t-t_t\right)a_o^2}{E^* L h^3}\right]^{0.25} \tag{4.18.76}$$

Using the calculated value of X_a enter either Table 4.18.3 or Figure 4.18.6 to determine Z_d, Z_v, Z_w, and Z_m.

d) STEP 4 – Calculate the following parameters

$$K = \frac{A}{D_o} \tag{4.18.77}$$

$$F = \frac{\left(1-\nu^*\right)\left(\lambda_s+\lambda_c+E\ln K\right)}{E^*} \tag{4.18.78}$$

$$\Phi = \left(1+\nu^*\right)F \tag{4.18.79}$$

$$Q_1 = \frac{\rho_s - 1 - \Phi\ Z_v}{1+\Phi\ Z_m} \tag{4.18.80}$$

$$Q_{Z1} = \frac{\left(Z_d + Q_1 \; Z_w\right) \; X_a^4}{2} \tag{4.18.81}$$

$$Q_{Z2} = \frac{\left(Z_v + Q_1 \; Z_m\right) \; X_a^4}{2} \tag{4.18.82}$$

$$U = \frac{\left[Z_w + \left(\rho_s - 1\right) \; Z_m\right] \; X_a^4}{1 + \Phi \; Z_m} \tag{4.18.83}$$

e) STEP 5 – Calculate the following quantities.

1) γ for Loading Cases 4, 5, 6, and 7. For Loading Cases 1, 2, and 3, $\gamma = 0$.

$$\gamma = \left[\alpha_{t,m}\left(T_{t,m} - T_a\right) - \alpha_{s,m}\left(T_{s,m} - T_a\right)\right]L \tag{4.18.84}$$

2) ω_s, $\omega_s{*}$, ω_c, and $\omega_c{*}$.

$$\omega_s = \rho_s k_s \beta_s \delta_s \left(1 + h\beta_s\right) \tag{4.18.85}$$

$$\omega_s{*} = \frac{a_o^2\left(\rho_s^2 - 1\right)\left(\rho_s - 1\right)}{4} - \omega_s \tag{4.18.86}$$

$$\omega_c = \rho_c k_c \beta_c \delta_c \left(1 + h\beta_c\right) \tag{4.18.87}$$

$$\omega_c{*} = a_o^2\left[\frac{\left(\rho_c^2 + 1\right)\left(\rho_c - 1\right)}{4} - \frac{\left(\rho_s - 1\right)}{2}\right] - \omega_c \tag{4.18.88}$$

3) γ_b

$$\gamma_b = 0 \qquad \textit{Configuration} - a \tag{4.18.89}$$

$$\gamma_b = \frac{G_c - C}{D_o} \qquad \textit{Configuration} - b \tag{4.18.90}$$

$$\gamma_b = \frac{G_c - G_1}{D_o} \qquad \textit{Configuration} - c \tag{4.18.91}$$

$$\gamma_b = \frac{G_c - G_s}{D_o} \qquad \textit{Configuration} - d \tag{4.18.92}$$

f) STEP 6 – For each loading case, calculate the effective pressure.

$$P_e = \frac{JK_{s,t}\left(P_s^{'} - P_t^{'} + P_\gamma + P_W + P_{\text{rim}}\right)}{1 + JK_{s,t}\left[Q_{Z1} + \left(\rho_s - 1\right)Q_{Z2}\right]} \tag{4.18.93}$$

where

$$P_s^{'} = \left(x_s + 2\left(1 - x_s\right)\nu_t + \frac{2}{K_{s,\,t}}\left(\frac{D_s}{D_o}\right)^2 \nu_s - \frac{\rho_s^2 - 1}{JK_{s,\,t}} - \frac{\left(1 - J\right)}{2JK_{s,\,t}}\frac{\left[D_J^2 - D_s^{\,2}\right]}{D_o^2}\right)P_s \tag{4.18.94}$$

$$P_t^{'} = \left(x_t + 2\left(1 - x_t\right)\nu_t + \frac{1}{JK_{s,t}}\right)P_t \tag{4.18.95}$$

$$P_\gamma = \frac{N_t K_t \gamma}{\pi\ a_o^2} \tag{4.18.96}$$

$$P_W = -\frac{U\gamma_b W}{2\pi a_o^2} \tag{4.18.97}$$

$$P_{\text{rim}} = -\frac{U\left(\omega_s * P_s - \omega_c * P_t\right)}{a_o^2} \tag{4.18.98}$$

g) STEP 7 – For each loading case check the bending stress.

1) Calculate Q_2

$$Q_2 = \frac{\left(\ \omega_s * P_s - \omega_c * P_t\ \ \right) + \dfrac{\gamma_b W}{2\pi}}{1 + \Phi\ Z_m} \tag{4.18.99}$$

2) Calculate the tubesheet bending stress

i) If $P_e \neq 0$, calculate Q_3

$$Q_3 = Q_1 + \frac{2\ Q_2}{P_e a_o^2} \tag{4.18.100}$$

For each loading case, determine coefficient F_m from either Table 4.18.3 or Figures 4.18.7 and 4.18.8. Calculate the tubesheet maximum bending stress

$$\sigma = \left(\frac{1.5\ F_m}{\mu *}\right)\left(\frac{2\ a_o}{h - h_g^{'}}\right)^2 P_e \tag{4.18.101}$$

ii) If $P_e = 0$, calculate the tubesheet maximum bending stress

$$\sigma = \frac{6Q_2}{\mu^* \left(h - h_g^{'}\right)^2} \tag{4.18.102}$$

3) Acceptance criteria

For Loading Cases 1, 2, and 3, if $|\sigma| \le 1.5S$, and for Loading Cases 4, 5, 6, and 7, if $|\sigma| \le S_{PS}$, the assumed tubesheet thickness is acceptable for bending. Otherwise, increase the assumed tubesheet thickness h and return to STEP 1.

h) STEP 8 – For each loading case, calculate the average shear stress in the tubesheet at the outer edge of the perforated region.

1) Calculate the average shear stress. The shear stress may be conservatively calculated using the following equation.

$$\tau = \left(\frac{1}{2\mu}\right)\left(\frac{a_o}{h}\right)P_e \tag{4.18.103}$$

If $|P_e| > \dfrac{3.2S\mu h}{D_o}$, the shear stress may be more accurately calculated by the following equation.

$$\tau = \left(\frac{1}{4\mu}\right)\left(\frac{1}{h}\left\{\frac{4A_p}{C_p}\right\}\right) \cdot P_e \tag{4.18.104}$$

2) Acceptance criteria

If $|\tau| \le 0.8S$, the assumed tubesheet thickness is acceptable for shear. Otherwise, increase the assumed tubesheet thickness h and return to STEP 1.

i) STEP 9 – Check the tube stress and tube-to-tubesheet joint design for each loading case.

1) Check the axial tube stress.

i) For each loading case, determine coefficients $F_{t,\min}$ and $F_{t,\max}$ from Table 4.18.4 and calculate the two extreme values of tube stress, $\sigma_{t,1}$ and $\sigma_{t,2}$. $\sigma_{t,1}$ and $\sigma_{t,2}$ may be positive or negative.

When $P_e \ne 0$

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[\left(P_s\, x_s - P_t\, x_t\right) - P_e\, F_{t,\min}\right] \tag{4.18.105}$$

$$\sigma_{t,2} = \frac{1}{x_t - x_s}\left[\left(P_s\, x_s - P_t\, x_t\right) - P_e\, F_{t,\max}\right] \tag{4.18.106}$$

When $P_e = 0$

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[\left(P_s\, x_s - P_t\, x_t\right) - \frac{2Q_2}{a_0^2}\, F_{t,\min}\right] \tag{4.18.107}$$

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[\left(P_s\ x_s - P_t\ x_t\right) - \frac{2Q_2}{a_0^2}\ F_{t,\max}\right] \tag{4.18.108}$$

ii) Determine $\sigma_{t,\max}$

$$\sigma_{t,\max} = \max\left[\left|\sigma_{t,1}\right|, \left|\sigma_{t,2}\right|\right] \tag{4.18.109}$$

2) Acceptance criteria

For Loading Cases 1, 2, and 3, if $\sigma_{t,\max} > S_t$, and for Loading Cases 4, 5, 6, and 7, if $\sigma_{t,\max} > 2S_t$, reconsider the tube design and return to STEP 1.
Otherwise, proceed to paragraph 4.18.8.4.i.3.

3) Check the tube-to-tubesheet joint design.

i) Calculate the largest tube-to-tubesheet joint load, W_t

$$W_t = \sigma_{t,\max}\pi\left(d_t - t_t\right)t_t \tag{4.18.110}$$

ii) Determine the maximum allowable load for the tube-to-tubesheet joint design, $L_{\max}$. For tube-to-tubesheet joints with full strength welds. $L_{\max}$ shall be determined in accordance with 4.18.10. For tube-to-tubesheet joints with partial strength welds, $L_{\max}$ shall be determined in accordance with 4.18.10 or Annex 4.C, as applicable. For all other tube joints, $L_{\max}$ shall be determined in accordance with Annex 4.C.

iii) Acceptance criteria

If $W_t > L_{\max}$, tube-to-tubesheet joint design shall be reconsidered.

If $W_t \le L_{\max}$, tube-to-tubesheet joint design is acceptable. Proceed to paragraph 4.18.8.4.i.4.

4) If $\sigma_{t,1}$ or $\sigma_{t,2}$ is negative, proceed to paragraph 4.18.8.4.i.6.

5) If $\sigma_{t,1}$ and $\sigma_{t,2}$ are positive, the tube design is acceptable. Proceed to STEP 10.

6) Check the tubes for buckling.

i) Calculate the largest equivalent unsupported buckling length of the tube l_t considering the unsupported tube spans l and their corresponding method of support defined by the parameter k.

$$l_t = kl \tag{4.18.111}$$

ii) Determine the maximum permissible buckling stress limit for the tubes.

$$S_{tb} = \min\left[\left\{\frac{\pi^2 E_t}{F_s F_t^2}\right\}, S_t\right] \qquad C_t \le F_t \tag{4.18.112}$$

$$S_{tb} = \min\left[\left\{\frac{S_{y,t}}{F_s}\left(1-\frac{F_t}{2\ C_t}\right)\right\},\ S_t\right] \qquad C_t > F_t \tag{4.18.113}$$

where

$$C_t = \sqrt{\frac{2\ \pi^2 E_t}{S_{y,t}}} \tag{4.18.114}$$

$$F_t = \frac{l_t}{r_t} \tag{4.18.115}$$

$$r_t = \frac{\sqrt{d_t^2 + \left(d_t - 2\ t_t\right)^2}}{4} \tag{4.18.116}$$

When $P_e \neq 0$

$$F_s = \min\left\{\max\left[\left(3.25 - 0.25\left(Z_d + Q_3 Z_w\right) X_a^4\right), 1.25\right], 2.0\right\} \tag{4.18.117}$$

When $P_e = 0$

$$F_s = 1.25 \tag{4.18.118}$$

iii) Determine $\sigma_{t,\min}$

$$\sigma_{t,\min} = \min\left[\sigma_{t,1}, \sigma_{t,2}\right] \tag{4.18.119}$$

iv) Acceptance criteria

If $\left|\sigma_{t,\min}\right| > S_{tb}$, reconsider the tube design and return to STEP 1.

If $\left|\sigma_{t,\min}\right| \leq S_{tb}$ the tube design is acceptable. Proceed to STEP 10.

j) STEP 10 –Perform this STEP for each loading case.

1) Calculate the axial membrane stress $\sigma_{s,m}$ in each different shell section. For shell sections integral with the tubesheet having a different material and/or thickness than the shell, refer to paragraph 4.18.8.5.

$$\sigma_{s,m} = \frac{a_o^2\left[P_e + \left(\rho_s^2 - 1\right)\left(P_s - P_t\right)\right]}{\left(D_s + t_s\right)t_s} + \frac{a_s^2 P_t}{\left(D_s + t_s\right)t_s} \tag{4.18.120}$$

2) Acceptance Criteria

i) For loading cases 1, 2,and 3, if $\left|\sigma_{s,m}\right| > S_s E_{s,w}$, and for loading cases 4,5,6, and 7, if $\left|\sigma_{s,m}\right| > S_{PS,s}$ reconsider the shell design and return to STEP 1.

ii) If $\sigma_{s,m}$ is negative, proceed to 4.18.8.4.j.3 below.

iii) If $\sigma_{s,m}$ is positive, the shell design is acceptable. Configurations a,b,and c: Proceed to STEP 11. Configuration d: the calculation procedure is complete.

3) Determine the maximum allowable longituidinal compressive stress, $S_{s,b}$

i) If $\left|\sigma_{s,m}\right| > S_{s,b}$, reconsider the shell design and return to STEP 1

ii) If $\left|\sigma_{s,m}\right| \le S_{s,b}$, the shell design is acceptable. Configuration a,b, and c: Proceed to STEP 11. Configuration d: The calculation procedure is complete.

k) STEP 11 – For each loading case, check the stresses in the shell and/or channel when integral with the tubesheet (Configurations a, b, and c).

1) Shell Stresses (Configurations a, b and c) – The shell shall have a uniform thickness of t_s for a minimum length of $1.8\sqrt{D_s\ t_s}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{s,m}$, the bending stress, $\sigma_{s,b}$,and total axial stress, σ_s, in the shell at its junction to the tubesheet.

$$\sigma_{s,m} = \frac{a_o^2\left[P_e + \left(\rho_s^2 - 1\right)\left(P_s - P_t\right)\right]}{\left(D_s + t_s\right)t_s} + \frac{a_s^2 P_t}{\left(D_s + t_s\right)t_s} \tag{4.18.121}$$

$$\sigma_{s,b} = \frac{6k_s}{t_s^2}\left[\beta_s\delta_s P + \frac{6\left(1-\left(\nu^*\right)^2\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1+\frac{h\beta_s}{2}\right)H\right] \tag{4.18.122}$$

$$H = P_e\left(Z_v + Z_m Q_1\right) + \frac{2Z_m Q_2}{a_o^2} \tag{4.18.123}$$

$$\sigma_s = \left|\sigma_{s,m}\right| + \left|\sigma_{s,b}\right| \tag{4.18.124}$$

2) Channel Stresses (Configuration a) – When the channel is cylindrical, it shall have a uniform thickness of t_c for a minimum length of $1.8\sqrt{D_c\ t_c}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{c,m}$, the bending stress, $\sigma_{c,b}$, and total axial stress, σ_c, in the channel at its junction to the tubesheet.

$$\sigma_{c,m} = \frac{a_c^2 P_t}{\left(D_c + t_c\right)t_c} \tag{4.18.125}$$

$$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c\delta_c P_t - \frac{6\left(1-\left(\nu^*\right)^2\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1+\frac{h\beta_c}{2}\right)H\right] \tag{4.18.126}$$

$$\sigma_c = \left|\sigma_{c,m}\right| + \left|\sigma_{c,b}\right| \tag{4.18.127}$$

3) Acceptance Criteria

i) Configuration a – For Loading Cases 1, 2, and 3, if $\sigma_s \le 1.5S_s$ and $\sigma_c \le 1.5S_c$, and for Loading Cases 4, 5, 6, and 7, if $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$, the shell and channel designs are acceptable and the calculation procedure is complete. Otherwise, proceed to STEP 12.

ii) Configurations b and c – For Loading Cases 1, 2, and 3, if $\sigma_s \le 1.5S_s$, and for Loading Cases 4, 5, 6, and 7, if $\sigma_s \le S_{PS,s}$, the shell design is acceptable and the calculation procedure is complete. Otherwise, proceed to STEP 12.

l) STEP 12 – The tubesheet design shall be reconsidered. One or a combination of the following three options may be used.

1) Option 1 – Increase the assumed tubesheet thickness h and return to STEP 1.

2) Option 2 – Increase the integral shell and/or channel thickness as follows and return to STEP 1.

i) Configurations a, b and c – If $\sigma_s > 1.5S_s$, increase the shell thickness t_s. It is permissible to increase the shell thickness adjacent to the tubesheet only (see Figure 4.18.9.)

ii) Configuration a – If $\sigma_c > 1.5S_c$, increase the channel thickness t_c.

3) Option 3 – Perform the elastic-plastic calculation procedure as defined in paragraph 4.18.8.6 only when the conditions of applicability stated in paragraph 4.18.8.6.b are satisfied.

4.18.8.5 Calculation Procedure for Effect of Different Shell Material and Thickness Adjacent to the Tubesheet

a) Scope

1) This procedure describes how to use the rules of paragraph 4.18.8.4 when the shell has a different thickness and/or a different material adjacent to the tubesheet (see Figure 4.18.9).

2) Use of this procedure may result in a smaller tubesheet thickness and should be considered when optimization of the tubesheet thickness or shell stress is desired.

b) Conditions of Applicability – This calculation procedure applies only when the shell is integral with the tubesheet (Configurations a, b, and c).

c) Calculation Procedure – The calculation procedure outlined in paragraph 4.18.8.4 shall be performed with the following modifications.

1) The shell shall have a thickness of $t_{s,1}$ for a minimum length of $1.8\sqrt{D_s t_{s,1}}$ adjacent to the tubesheets.

2) In STEP 2, replace the equation for K_s with,

$$K_s^* = \frac{\pi\left(D_s + t_s\right)}{\left(\dfrac{L - l_1 - l_1'}{E_s t_s}\right) + \left(\dfrac{l_1 + l_1'}{E_{s,1} t_{s,1}}\right)} \tag{4.18.128}$$

Calculate $K_{s,t}$ and J, replacing K_s with K_s^*, and calculate β_s, k_s, and δ_s, replacing t_s with $t_{s,1}$ and E_s with $E_{s,1}$.

3) In STEP 5, replace the equation for γ with.

$$\gamma^* = \left(T_{t,m} - T_a\right)\alpha_{t,m}L - \left(T_{s,m} - T_a\right)\left[\alpha_{s,m}\left(L - l_1 - l_1^{'}\right) + \alpha_{s,m,1}\left(l_1 + l_1^{'}\right)\right] \tag{4.18.129}$$

4) In STEP 6, calculate P_γ, replacing γ with γ^*.

5) In STEP 10, calculate $\sigma_{s,m}$, replacing t_s with $t_{s,1}$. Replace S with $S_{s,1}$, and $S_{s,b}$ with $S_{s,b,1}$.

6) In STEP 11, calculate $\sigma_{s,m}$ and $\sigma_{s,b}$, replacing t_s with $t_{s,1}$, and E with $E_{s,1}$. Replace S_s with $S_{s,1}$, and $S_{PS,s}$ with $S_{SP,s,1}$.

7) If the elastic-plastic calculation procedure of paragraph 4.18.8.6 is being performed, replace $S_{y,s}$ with $S_{y,s,1}$, $S_{PS,s}$ with $S_{SP,s,1}$, and E with $E_{s,1}$ in this calculation.

8) If the radial thermal expansion procedure of paragraph 4.18.8.7 is being performed, replace t_s with $t_{s,1}$, and E with $E_{s,1}$ in this calculation.

4.18.8.6 Calculation Procedure for Effect of Plasticity at Tubesheet/Channel or Shell Joint

a) Scope – This procedure describes how to use the rules of paragraph 4.18.8.4 when the effect of plasticity at the shell-tubesheet and/or channel-tubesheet joint is to be considered.

1) If the discontinuity stresses at the shell-tubesheet and/or channel-tubesheet joint exceed the allowable stress limits, the thickness of the shell, channel, or tubesheet may be increased to meet the stress limits given in paragraph 4.18.8.4. As an alternative, when the calculated tubesheet stresses are within the allowable stress limits, but either or both of the calculated shell or channel total stresses exceed their allowable stress limits, one additional "elastic-plastic solution" calculation may be performed.

2) This calculation permits a reduction of the shell and/or channel modulus of elasticity, where it affects the rotation of the joint, to reflect the anticipated load shift resulting from plastic action at the joint. The reduced effective modulus has the effect of reducing the shell and/or channel stresses in the elastic-plastic calculation; however, due to load shifting this usually leads to an increase in the tubesheet stress. In most cases, an elastic-plastic calculation using the appropriate reduced shell or channel modulus of elasticity results in a design where the calculated tubesheet stresses are within the allowable stress limits.

b) Conditions of Applicability

1) This procedure shall not be used at temperatures where the time-dependent properties govern the allowable stress.

2) This procedure applies only for Loading Cases 1, 2, and 3.

3) This procedure applies to Configuration a when $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$.

4) This procedure applies to Configurations b and c when $\sigma_s \le S_{PS,s}$.

5) This procedure may only be used once for each iteration of tubesheet, shell, and channel thickness and change of materials.

c) Calculation Procedure – After the calculation procedure given in paragraph 4.18.8.4 (STEP 1 through 11) has been performed for the elastic solution, an elastic-plastic calculation using the referenced steps from paragraph 4.18.8.4 shall be performed in accordance with the following procedure for each

applicable loading case. Except for those quantities modified below, the quantities to be used for the elastic-plastic calculation shall be the same as those calculated for the corresponding elastic loading case.

1) Define the maximum permissible bending stress limit in the shell and channel.

$$S_s{}^* = \min\left[S_{y,s}, \left(\frac{S_{PS,s}}{2}\right)\right] \qquad Configurations - a,b,c \qquad (4.18.130)$$

$$S_c{}^* = \min\left[S_{y,c}, \left(\frac{S_{PS,c}}{2}\right)\right] \qquad Configuration - a \qquad (4.18.131)$$

2) Using bending stresses $\sigma_{s,b}$ and $\sigma_{c,b}$ calculated in STEP 11 for the elastic solution, determine $fact_s$ and $fact_c$ as follows.

$$fact_s = \min\left[\left(1.4 - \frac{0.4|\sigma_{s,b}|}{S_s{}^*}\right), 1.0\right] \qquad Configurations - a,b,c \qquad (4.18.132)$$

$$fact_c = \min\left[\left(1.4 - \frac{0.4|\sigma_{c,b}|}{S_c{}^*}\right), 1.0\right] \qquad Configuration - a \qquad (4.18.133)$$

3) For Configuration a, if $fact_s = 1.0$ and $fact_c = 1.0$, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to paragraph 4.18.8.6.c.4. For Configurations b and c, if $fact_s = 1.0$, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to paragraph 4.18.8.6.c.4.

4) Calculate reduced values of E_s and E_c as follows:

$$E_s{}^* = E_s \cdot fact_s \qquad Configurations - a,b,c \qquad (4.18.134)$$

$$E_c{}^* = E_c \cdot fact_c \qquad Configuration - a \qquad (4.18.135)$$

5) In STEP 2, recalculate k_s and λ_s by replacing E_s with $E_s{}^*$, and k_c and λ_c by replacing E_c with $E_c{}^*$.

6) In STEP 4, recalculate F, Φ, Q_1, Q_{Z1}, Q_{Z2}, , and U.

7) In STEP 6, recalculate P_W, P_{rim}, and P_e.

8) In STEP 7, recalculate Q_2, Q_3, F_m, and the tubesheet bending stress σ. If $|\sigma| \le 1.5S$, the design is acceptable and the calculation procedure is complete. Otherwise, the unit geometry shall be reconsidered.

4.18.8.7 Calculation Procedure for Effect of Radial Differential Thermal Expansion Adjacent to the Tubesheet

a) Scope

1) This procedure describes how to use the rules of paragraph 4.18.8.4 when the effect of radial differential thermal expansion between the tubesheet and integral shell or channel is to be considered.

2) This procedure shall be used when cyclic or dynamic reactions due to pressure or thermal variations are specified.

3) This procedure shall be used when specified by the user. The user shall provide the Manufacturer with the data necessary to determine the required tubesheet, channel, and shell metal temperatures.

4) Optionally, the designer may use this procedure to consider the effect of radial differential thermal expansion even when it is not required by paragraphs 4.18.8.7.a.2 or 4.18.8.7.a.3.

b) Conditions of Applicability – This calculation procedure applies only when the tubesheet is integral with the shell or channel (Configurations a, b, and c).

c) Calculation Procedure – The calculation procedure outlined in paragraph 4.18.8.4 and 4.18.8.5, if applicable, shall be performed only for Loading Cases 4, 5, 6 and 7, accounting for the following modifications.

1) Determine the average temperature of the unperforated rim T_r.

$$T_r = \frac{T' + T_s' + T_c'}{3} \qquad Configuration - a \qquad (4.18.136)$$

$$T_r = \frac{T' + T_s'}{2} \qquad Configurations - b, c \qquad (4.18.137)$$

For conservative values of P_s* and P_c*, $T_r = T'$ may be used.

2) Determine the average temperature of the shell T_s* and channel T_c* at their junction to the tubesheet using the equations shown below.

$$T_s* = \frac{T_s' + T_r}{2} \qquad Configurations - a, b, c \qquad (4.18.138)$$

$$T_c* = \frac{T_c' + T_r}{2} \qquad Configuration - a \qquad (4.18.139)$$

For conservative values of P_s* and P_c*, $T_s* = T_s'$ and $T_c* = T_c'$ may be used.

3) Calculate P_s* and P_c*.

$$P_s* = \frac{E_s t_s \left[\alpha_s' \left(T_s* - T_a \right) - \alpha' \left(T_r - T_a \right) \right]}{a_s} \qquad Configurations - a, b, c \qquad (4.18.140)$$

$$P_c^* = \frac{E_c t_c \left[\alpha_c^{'} \left(T_c^* - T_a \right) - \alpha^{'} \left(T_r - T_a \right) \right]}{a_c} \qquad Configuration - a \tag{4.18.141}$$

$$P_c^* = 0.0 \qquad Configurations - b, c \tag{4.18.142}$$

4) Calculate P_ω.

$$P_\omega = \frac{U \left(\omega_s P_s^* - \omega_c P_c^* \right)}{a_o^2} \tag{4.18.143}$$

5) In STEP 6, replace the equation for P_e with:

$$P_e = \frac{J K_{s,t} \left(P_s^{'} - P_t^{'} + P_\gamma + P_\omega + P_W + P_{\text{rim}} \right)}{1 + J K_{s,t} \left[Q_{Z1} + \left(\rho_s - 1 \right) Q_{Z2} \right]} \tag{4.18.144}$$

6) In STEP 7, replace the equation for Q_2 with:

$$Q_2 = \frac{\left(\omega_s^* P_s - \omega_c^* P_t \right) - \left(\omega_s P_s^* - \omega_c P_c^* \right) + \frac{\gamma_b W}{2\pi}}{1 + \Phi Z_m} \tag{4.18.145}$$

7) In STEP 11, replace the equations for $\sigma_{s,b}$ and $\sigma_{c,b}$ with the following equations where H is given by Equation (4.18.122).

$$\sigma_{s,b} = \frac{6 k_s}{t_s^2} \left[\beta_s \left(\delta_s P_s + \frac{a_s^2 P_s^*}{E_s t_s} \right) + \frac{6 \left(1 - \nu^{*2} \right)}{E^*} \left(\frac{a_o^3}{h^3} \right) \left(1 + \frac{h \beta_s}{2} \right) H \right] \tag{4.18.146}$$

$$\sigma_{c,b} = \frac{6 k_c}{t_c^2} \left[\beta_c \left(\delta_c P_t + \frac{a_c^2 P_c^*}{E_c t_c} \right) - \frac{6 \left(1 - \nu^{*2} \right)}{E^*} \left(\frac{a_o^3}{h^3} \right) \left(1 + \frac{h \beta_c}{2} \right) H \right] \tag{4.18.147}$$

4.18.8.8 Calculation Procedure for Simply Supported Fixed Tubesheets

4.18.8.8.1 Scope

This procedure describes how to use the rules of 4.18.8.4 when the effect of the stiffness of the integral channel and/or shell is not considered.

4.18.8.8.2 Conditions of Applicability

This calculation applies only when the tubesheet is integral with the shell or channel (Configurations a,b, and c).

4.18.8.8.3 Calculation Procedure

The calculation procedure outlined in 4.18.8.4 shall be performed accounting for the following modifications;

a) Perform STEPs 1 through 10.

b) Perform STEP 11 except as follows:

1) The shell (Configuration a,b,and c) is not required to meet a minimum length requirement. The shell is exempt from the minimum length requirement in 4.18.8.5.c.1.

2) The channel (Configuration a) is not required to meet a minimum length requirement.

3) Acceptance Criteria

i) Configuration a: If $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$, then the shell and channel are acceptable. Otherwise increase the thickness of the overstressed component(s) (shell and/or channel) and return to STEP 1.

ii) Configuration b and c : If $\sigma_s \le S_{PS,s}$ then the shell is acceptable. Otherwise increase the thickness of the shell and return to STEP 1.

c) Do not perform STEP 12

d) Repeat STEPs 1 - 7 for loading cases 1 - 3, with the following changes to STEP 2 until the tubesheet stress criteria have been met:

1) Configurations a, b, and c: $\beta_s = 0,\ k_s = 0,\ \lambda_s = 0,\ \delta_s = 0$.

2) Configuration a: $\beta_c = 0,\ k_c = 0,\ \lambda_c = 0,\ \delta_c = 0$.

4.18.9 Rules for the Design of Floating Tubesheets

4.18.9.1 Scope

a) These rules cover the design of tubesheets for floating tubesheet heat exchangers that have one stationary tubesheet and one floating tubesheet. Three types of floating tubesheet heat exchangers are covered as shown in Figure 4.18.10.

1) Sketch (a), immersed floating head;

2) Sketch (b), externally sealed floating head;

3) Sketch (c), internally sealed floating tubesheet.

b) Stationary tubesheets may have one of the six configurations shown in Figure 4.18.11.

1) Configuration a: tubesheet integral with shell and channel;

2) Configuration b: tubesheet integral with shell and gasketed with channel, extended as a flange;

3) Configuration c: tubesheet integral with shell and gasketed with channel, not extended as a flange;

4) Configuration d: tubesheet gasketed with shell and channel, extended or not extended as a flange;

5) Configuration e: tubesheet gasketed with shell and integral with channel, extended as a flange;

6) Configuration f: tubesheet gasketed with shell and integral with channel, not extended as a flange.

c) Floating tubesheets may have one of the four configurations shown in Figure 4.18.12.

1) Configuration A: tubesheet integral;

2) Configuration B: tubesheet gasketed, extended as a flange;

3) Configuration C: tubesheet gasketed, not extended as a flange;

4) Configuration D: tubesheet internally sealed.

4.18.9.2 Conditions of Applicability

The two tubesheets shall have the same thickness and material.

4.18.9.3 Design Considerations

a) The calculation shall be performed for the stationary end and for the floating end of the exchanger. Since the edge configurations of the stationary and floating tubesheets are different, the data may be different for each set of calculations. However the conditions of applicability given in 4.18.9.2 must be maintained. For the stationary end, diameters A, C, D_s, D_c, G_s, G_c, G_1, and the thickness t_c shall be taken from Figure 4.18.11. For the floating end, diameters A, C, D_c, G_c, and the thickness t_c shall be taken from Figure 4.18.12, and the radial shell dimension a_s shall be taken equal to a_c

b) It is generally not possible to determine, by observation, the most severe condition of coincident pressure, temperature and radial differential thermal expansion. Thus, it is necessary to evaluate all the anticipated loading conditions to ensure that the worst load combination has been considered in the design. The various loading conditions to be considered shall include the normal operating conditions, the startup conditions, the shutdown conditions, and the upset conditions, which may govern the design of the main components of the heat exchanger (i.e., tubesheets, tubes, shell, channel, tube-to-tubesheet joint).

1) For each of these conditions, the following loading cases shall be considered to determine the effective pressure P_e to be used in the design equations:

i) Loading Case 1 – Tube side pressure P_t acting only $\left(P_s = 0\right)$ without differential thermal expansion.

ii) Loading Case 2 – Shell side pressure P_s acting only $\left(P_t = 0\right)$ without differential thermal expansion.

iii) Loading Case 3 – Tube side pressure P_t and shell side pressure P_s acting simultaneously, without differential thermal expansion.

iv) Loading Case 4 – Radial differential thermal expansion acting only $\left(P_s = 0\right)$ and $\left(P_t = 0\right)$.

v) Loading Case 5 – Tube side pressure P_t acting only $\left(P_s = 0\right)$ with radial differential thermal expansion.

vi) Loading Case 6 – Shell side pressure P_s acting only $\left(P_t = 0\right)$ with radial differential thermal expansion.

vii) Loading Case 7 – Tube side pressure P_t and shell side pressure P_s acting simultaneously, with radial differential thermal expansion.

2) Loading Cases 4, 5, 6, and 7 are only required when the effect of radial differential thermal expansion is to be considered.

3) When vacuum exists, each loading case shall be considered with and without the vacuum.

4) When differential design pressure is specified by the user, the design shall be based only on Loading Cases 3 and 7. If the tube side is the higher-pressure side, P_t shall be the tube side design pressure and P_s shall be P_t less the differential design pressure. If the shell side is the higher-pressure side, P_s shall be the shell side design pressure and P_t shall be P_s less the differential design pressure.

5) The designer should take appropriate consideration of the stresses resulting from the pressure test required by paragraph 4.1.6.2 and Part 8.

c) Elastic moduli, yield strengths, and allowable stresses shall be taken at design temperatures. However for cases involving thermal loading (Loading Cases 4, 5, 6, and 7), it is permitted to use the operating temperatures instead of the design temperatures.

d) As the calculation procedure is iterative, a value h shall be assumed for the tubesheet thickness to calculate and check that the maximum stresses in tubesheet, tubes, shell, and channel are within the maximum permissible stress limits and that the resulting tube-to-tubesheet joint load is acceptable.

e) The designer shall consider the following:

   1) The effect of deflections in the tubesheet design, especially when the tubesheet thickness h is less than the tube diameter.

   2) The effect of radial differential thermal expansion between the tubesheet and integral shell or channel (Configurations a, b, c, e, f, and A) in accordance with paragraph 4.18.9.5.

f) The designer may consider the tubesheet as simply supported in accordance with 4.18.9.6.

4.18.9.4 Calculation Procedure

The procedure for the design of tubesheets for a floating tubesheet heat exchanger is as follows. Calculations shall be performed for both the stationary tubesheet and the floating tubesheet.

a) STEP 1 – Determine D_o, μ, μ^*, and h'_g from paragraph 4.18.6.4.a. For Loading Cases 4, 5, 6, and 7, $h'_g = 0$. Calculate the following quantities.

$$a_o = \frac{D_o}{2} \tag{4.18.148}$$

$$a_s = \frac{D_s}{2} \qquad Configurations - a,b,c \tag{4.18.149}$$

$$a_s = \frac{G_s}{2} \qquad Configurations - d,e,f \tag{4.18.150}$$

$$a_s = a_c \qquad Configurations - A,B,C,D \tag{4.18.151}$$

$$a_c = \frac{D_c}{2} \qquad Configurations - a,e,f,A \tag{4.18.152}$$

$$a_c = \frac{G_c}{2} \qquad Configurations - b,c,d,B,C \tag{4.18.153}$$

$$a_c = \frac{A}{2} \qquad Configuration - D \tag{4.18.154}$$

$$\rho_s = \frac{a_s}{a_o} \tag{4.18.155}$$

$$\rho_c = \frac{a_c}{a_o} \tag{4.18.156}$$

$$x_s = 1 - N_t \left(\frac{d_t}{2\ a_o} \right)^2 \tag{4.18.157}$$

$$x_t = 1 - N_t \left(\frac{d_t - 2t_t}{2\ a_o} \right)^2 \tag{4.18.158}$$

b) STEP 2 – Calculate the shell and channel coefficients.

1) The shell coefficients for Configurations a, b and c.

$$\beta_s = \frac{\left[12\left(1-\nu_s^2\right)\right]^{0.25}}{\left[\left(D_s + t_s\right)t_s\right]^{0.5}} \tag{4.18.159}$$

$$k_s = \frac{\beta_s E_s t_s^3}{6\left(1-\nu_s^2\right)} \tag{4.18.160}$$

$$\lambda_s = \frac{6 D_s k_s}{h^3}\left(1 + h\beta_s + \frac{h^2 \beta_s^2}{2}\right) \tag{4.18.161}$$

$$\delta_s = \frac{D_s^2}{4 E_s t_s}\left(1 - \frac{\nu_s}{2}\right) \tag{4.18.162}$$

For Configurations d, e , f, A, B, C, and D, $\beta_s = k_s = \lambda_s = \delta_s = 0$.

2) The channel coefficients for Configurations a, e, f, and A.

$$\beta_c = \frac{\left[12\ \left(1-\nu_c^2\right)\right]^{0.25}}{\left[\left(D_c + t_c\right)t_c\right]^{0.5}} \tag{4.18.163}$$

$$k_c = \frac{\beta_c E_c t_c^3}{6\left(1-\nu_c^2\right)} \tag{4.18.164}$$

$$\lambda_c = \frac{6 D_c k_c}{h^3}\left(1 + h\beta_c + \frac{h^2 \beta_c^2}{2}\right) \tag{4.18.165}$$

$$\delta_c = \frac{D_c^2}{4 E_c t_c}\left(1 - \frac{\nu_c}{2}\right) \qquad \textit{For a Cylinder} \tag{4.18.166}$$

$$\delta_c = \frac{D_c^2}{4E_c t_c}\left(\frac{1-\nu_c}{2}\right) \qquad \textit{For a Hemispherical Head} \tag{4.18.167}$$

For Configurations b, c, d, B, C, and D, $\beta_c = k_c = \lambda_c = \delta_c = 0$.

c) STEP 3 – Calculate h/p. Determine E^*/E and ν^* using paragraph 4.18.6.4.b. Calculate X_a.

$$X_a = \left[\frac{24\left(1-(\nu^*)^2\right)N_t E_t t_t\left(d_t - t_t\right)a_o^2}{E^* L h^3}\right]^{0.25} \tag{4.18.168}$$

Using the calculated value of X_a enter either Table 4.18.3 or Figure 4.18.6 to determine Z_d, Z_v, Z_w, and Z_m.

d) STEP 4 – Calculate the following parameters.

$$K = \frac{A}{D_o} \tag{4.18.169}$$

$$F = \frac{(1-\nu^*)\left(\lambda_s + \lambda_c + E\ln K\right)}{E^*} \tag{4.18.170}$$

$$\Phi = (1+\nu^*)F \tag{4.18.171}$$

$$Q_1 = \frac{\rho_s - 1 - \Phi\, Z_v}{1 + \Phi\, Z_m} \tag{4.18.172}$$

e) STEP 5 – Calculate the following quantities.

1) ω_s, ω_s^*, ω_c, and ω_c^*.

$$\omega_s = \rho_s k_s \beta_s \delta_s \left(1 + h\beta_s\right) \tag{4.18.173}$$

$$\omega_s^* = \frac{a_o^2\left(\rho_s^2 - 1\right)\left(\rho_s - 1\right)}{4} - \omega_s \tag{4.18.174}$$

$$\omega_c = \rho_c k_c \beta_c \delta_c \left(1 + h\beta_c\right) \tag{4.18.175}$$

$$\omega_c^* = a_o^2\left[\frac{\left(\rho_c^2 + 1\right)\left(\rho_c - 1\right)}{4} - \frac{\left(\rho_s - 1\right)}{2}\right] - \omega_c \tag{4.18.176}$$

2) γ_b.

$$\gamma_b = 0.0 \qquad \textit{Configuration} - a, A, D \tag{4.18.177}$$

$$\gamma_b = \frac{G_c - C}{D_o} \qquad Configuration - b, B \tag{4.18.178}$$

$$\gamma_b = \frac{G_c - G_1}{D_o} \qquad Configuration - c, C \tag{4.18.179}$$

$$\gamma_b = \frac{G_c - G_s}{D_o} \qquad Configuration - d \tag{4.18.180}$$

$$\gamma_b = \frac{C - G_s}{D_o} \qquad Configuration - e \tag{4.18.181}$$

$$\gamma_b = \frac{G_1 - G_s}{D_o} \qquad Configuration - f \tag{4.18.182}$$

f) STEP 6 – For each loading case, calculate the effective pressure P_e.

1) For an exchanger with an immersed floating head (see Figure 4.18.10 Sketch (a)):

$$P_e = P_s - P_t \tag{4.18.183}$$

2) For an exchanger with an externally sealed floating head (see Figure 4.18.10 Sketch (b)):

$$P_e = P_s\left(1 - \rho_s^2\right) - P_t \tag{4.18.184}$$

3) For an exchanger with an internally sealed floating tubesheet (see Figure 4.18.10 Sketch (c)):

$$P_e = \left(P_s - P_t\right)\left(1 - \rho_s^2\right) \tag{4.18.185}$$

g) STEP 7 – For each loading case check the bending stress.

1) Calculate Q_2.

$$Q_2 = \frac{\left(\omega_s * P_s - \omega_c * P_t\right) + \dfrac{\gamma_b W}{2\pi}}{1 + \Phi\, Z_m} \tag{4.18.186}$$

2) Calculate the tubesheet bending stress.

i) If $P_e \neq 0$, calculate Q_3.

$$Q_3 = Q_1 + \frac{2\,Q_2}{P_e a_o^2} \tag{4.18.187}$$

For each loading case, determine coefficient F_m from either Table 4.18.3 or Figures 4.18.7 and 4.18.8. Calculate the maximum tubesheet bending stress.

$$\sigma = \left(\frac{1.5\,F_m}{\mu^*}\right)\left(\frac{2\,a_o}{h-h_g^{'}}\right)^2 P_e \tag{4.18.188}$$

ii) If $P_e = 0$, calculate the maximum tubesheet bending stress.

$$\sigma = \frac{6\,Q_2}{\mu^*\left(h-h_g^{'}\right)^2} \tag{4.18.189}$$

3) Acceptance criteria

For Loading Cases 1, 2, and 3, if $|\sigma| \le 1.5S$, and for Loading Cases 4, 5, 6, and 7, if $|\sigma| \le S_{PS}$, the assumed tubesheet thickness is acceptable for bending. Otherwise, increase the assumed tubesheet thickness h and return to STEP 1.

For Configurations a, b, c, d, e, and f, proceed to STEP 8. For Configuration A, proceed to STEP 10. For Configurations B, C, and D, the calculation is complete.

h) STEP 8 – For each loading case, check the average shear stress in the tubesheet at the outer edge of the perforated region

1) Calculate the average shear stress. The shear stress may be conservatively calculated using the following equation.

$$\tau = \left(\frac{1}{2\mu}\right)\left(\frac{a_o}{h}\right)P_e \tag{4.18.190}$$

If $|P_e| > \dfrac{3.2S\mu h}{D_o}$, the shear stress may be more accurately calculated by the following equation.

$$\tau = \left(\frac{1}{4\mu}\right)\left(\frac{1}{h}\left\{\frac{4A_p}{C_p}\right\}\right)\cdot P_e \tag{4.18.191}$$

2) Acceptance criteria

If $|\tau| \le 0.8S$, the assumed tubesheet thickness is acceptable for shear. Otherwise, increase the assumed tubesheet thickness h and return to STEP 1.

i) STEP 9 – Check the tube stress and tube-to-tubesheet joint design for each loading case.

1) Check the axial tube stress.

i) For each loading case, determine the coefficients $F_{t,\min}$ and $F_{t,\max}$ from Table 4.18.4 and calculate the two extreme values of tube stress, $\sigma_{t,1}$ and $\sigma_{t,2}$. $\sigma_{t,1}$ and $\sigma_{t,2}$ may be positive or negative.

When $P_e \ne 0$

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[\left(P_s\,x_s - P_t\,x_t\right) - P_e\,F_{t,\min}\right] \tag{4.18.192}$$

$$\sigma_{t,2} = \frac{1}{x_t - x_s}\left[\left(P_s\ x_s - P_t\ x_t\right) - P_e\ F_{t,\max}\right] \tag{4.18.193}$$

When $P_e = 0$

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[\left(P_s\ x_s - P_t\ x_t\right) - \frac{2Q_2}{a_0^2}\ F_{t,\min}\right] \tag{4.18.194}$$

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[\left(P_s\ x_s - P_t\ x_t\right) - \frac{2Q_2}{a_0^2}\ F_{t,\max}\right] \tag{4.18.195}$$

ii) Determine $\sigma_{t,\max}$

$$\sigma_{t,\max} = \max\left[\left|\sigma_{t,1}\right|, \left|\sigma_{t,2}\right|\right] \tag{4.18.196}$$

2) Acceptance criteria

For Loading Cases 1, 2, and 3, if $\sigma_{t,\max} > S_t$, and for Loading Cases 4, 5, 6, and 7, if $\sigma_{t,\max} > 2S_t$, reconsider the design and return to STEP 1.
Otherwise, proceed to paragraph 4.18.9.4.i.3.

3) Check the tube-to-tubesheet joint design.

i) Calculate the largest tube-to-tubesheet joint load, W_t

$$W_t = \sigma_{t,\max}\pi\left(d_t - t_t\right)t_t \tag{4.18.197}$$

ii) Determine the maximum allowable load for the tube-to-tubesheet joint design, $L_{\max}$. For tube-to-tubesheet joints with full strength welds. $L_{\max}$ shall be determined in accordance with 4.18.10. For tube-to-tubesheet joints with partial strength welds, $L_{\max}$ shall be determined in accordance with 4.18.10 or Annex 4.C, as applicable. For all other tube joints, $L_{\max}$ shall be determined in accordance with Annex 4.C.

iii) Acceptance criteria

If $W_t > L_{\max}$, tube-to-tubesheet joint design shall be reconsidered.

If $W_t \le L_{\max}$, tube-to-tubesheet joint design is acceptable. Proceed to paragraph 4.18.9.4.i.4.

4) If $\sigma_{t,1}$ or $\sigma_{t,2}$ is negative, proceed to paragraph 4.18.9.4.i.6.

5) If $\sigma_{t,1}$ and $\sigma_{t,2}$ are positive, the tube design is acceptable. Proceed to STEP 10.

6) Check the tubes for buckling.

i) Calculate the largest equivalent unsupported buckling length of the tube l_t considering the unsupported tube spans l and their corresponding method of support defined by the parameter k.

$$l_t = kl \tag{4.18.198}$$

ii) Determine the maximum permissible buckling stress limit S_{tb} for the tubes.

$$S_{tb} = \min\left[\left\{\frac{\pi^2 E_t}{F_s F_t^2}\right\},\ S_t\right] \qquad C_t \le F_t \tag{4.18.199}$$

$$S_{tb} = \min\left[\left\{\frac{S_{y,t}}{F_s}\left(1-\frac{F_t}{2\ C_t}\right)\right\},\ S_t\right] \qquad C_t > F_t \tag{4.18.200}$$

where

$$F_q = \frac{\left(Z_d + Q_3\ Z_v\right)\ X_a^4}{2} \tag{4.18.201}$$

$$C_t = \sqrt{\frac{2\ \pi^2 E_t}{S_{y,t}}} \tag{4.18.202}$$

$$F_t = \frac{l_t}{r_t} \tag{4.18.203}$$

$$r_t = \frac{\sqrt{d_t^2 + \left(d_t - 2\ t_t\right)^2}}{4} \tag{4.18.204}$$

When $P_e \ne 0$

$$F_s = \min\left\{\max\left[\left(3.25 - 0.25\left(Z_d + Q_3 Z_w\right) X_a^4\right), 1.25\right], 2.0\right\} \tag{4.18.205}$$

When $P_e = 0$

$$F_s = 1.25 \tag{4.18.206}$$

iii) Determine $\sigma_{t,\min}$

$$\sigma_{t,\min} = \min\left[\sigma_{t,1}, \sigma_{t,2}\right] \tag{4.18.207}$$

iv) Acceptance criteria

If $\left|\sigma_{t,\min}\right| > S_{tb}$, reconsider the tube design and return to STEP 1.

If $\left|\sigma_{t,\min}\right| \le S_{tb}$, the tube design is acceptable. Proceed to STEP 10.

j) STEP 10 – For each loading case, check the stresses in the shell and/or channel integral with the tubesheet.

1) Configurations a, b and c – The shell shall have a uniform thickness of t_s for a minimum length of

$1.8\sqrt{D_s\ t_s}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{s,m}$, the bending stress, $\sigma_{s,b}$, and total axial stress, σ_s, in the shell at its junction to the tubesheet, where H is given by Equation (4.18.122).

$$\sigma_{s,m} = \frac{a_o^2\left[P_e + \left(\rho_s^2 - 1\right)\left(P_s - P_t\right)\right]}{\left(D_s + t_s\right)t_s} + \frac{a_s^2 P_t}{\left(D_s + t_s\right)t_s} \tag{4.18.208}$$

$$\sigma_{s,b} = \frac{6k_s}{t_s^2}\left[\beta_s\delta_s P_s + \frac{6\left(1-(\nu^*)^2\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1+\frac{h\beta_s}{2}\right)H\right] \tag{4.18.209}$$

$$\sigma_s = \left|\sigma_{s,m}\right| + \left|\sigma_{s,b}\right| \tag{4.18.210}$$

2) Configurations a, e, f, and A – A cylindrical channel shall have a uniform thickness of t_c for a minimum length of $1.8\sqrt{D_c\ t_c}$ adjacent to the tubesheet. Calculate the axial membrane stress, $\sigma_{c,m}$, the bending stress, $\sigma_{c,b}$, and total axial stress, σ_c, in the channel at its junction to the tubesheet, where H is given by Equation (4.18.122).

$$\sigma_{c,m} = \frac{a_c^2 P_t}{\left(D_c + t_c\right)t_c} \tag{4.18.211}$$

$$\sigma_{c,b} = \frac{6k_c}{t_c^2}\left[\beta_c\delta_c P_t - \frac{6\left(1-(\nu^*)^2\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1+\frac{h\beta_c}{2}\right)H\right] \tag{4.18.212}$$

$$\sigma_c = \left|\sigma_{c,m}\right| + \left|\sigma_{c,b}\right| \tag{4.18.213}$$

3) Acceptance Criteria

i) Configuration a – For Loading Cases 1, 2, and 3, if $\sigma_s \le 1.5S_s$ and $\sigma_c \le 1.5S_c$, and for Loading Cases 4, 5, 6, and 7, if $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$, the shell and channel designs are acceptable and the calculation procedure is complete. Otherwise, proceed to STEP 11.

ii) Configurations b and c – For Loading Cases 1, 2, and 3, if $\sigma_s \le 1.5S_s$, and for Loading Cases 4, 5, 6, and 7, if $\sigma_s \le S_{PS,s}$, the shell design is acceptable and the calculation procedure is complete. Otherwise, proceed to STEP 11.

iii) Configurations e, f, and A – For Loading Cases 1, 2, and 3, if $\sigma_c \le 1.5S_c$, and for Loading Cases 4, 5, 6, and 7, if $\sigma_c \le S_{PS,c}$, the channel design is acceptable and the calculation procedure is complete. Otherwise, proceed to STEP 11.

k) STEP 11 – The design shall be reconsidered. One or a combination of the following three options may be used.

1) Option 1 – Increase the assumed tubesheet thickness h and return to STEP 1.

2) Option 2 – Increase the integral shell and/or channel thickness and return to STEP 1.

   i) Configurations a, b and c – If $\sigma_s > 1.5S_s$, increase the shell thickness t_s.

   ii) Configurations a, e, f and A – If $\sigma_c > 1.5S_c$, increase the channel thickness t_c.

3) Option 3 - Perform the elastic-plastic calculation procedure as defined in 4.18.9.6 only when the conditions of applicability stated in 4.18.9.6.b are satisfied.

4.18.9.5 Calculation Procedure for Effect of Plasticity at Tubesheet/Channel or Shell Joint

a) Scope – This procedure describes how to use the rules of paragraph 4.18.9.4 when the effect of plasticity at the shell-tubesheet and/or channel-tubesheet joint is to be considered.

   1) When the calculated tubesheet stresses are within the allowable stress limits, but either or both of the calculated shell or channel total stresses exceed their allowable stress limits, an additional "elastic-plastic solution" calculation may be performed.

   2) This calculation permits a reduction of the shell and/or channel modulus of elasticity, where it affects the rotation of the joint, to reflect the anticipated load shift resulting from plastic action at the joint. The reduced effective modulus has the effect of reducing the shell and/or channel stresses in the elastic-plastic calculation; however, due to load shifting this usually leads to an increase in the tubesheet stress. In most cases, an elastic-plastic calculation using the appropriate reduced shell or channel modulus of elasticity results in a design where the calculated tubesheet stresses are within the allowable stress limits.

b) Conditions of Applicability

   1) This procedure shall not be used at temperatures where the time-dependent properties govern the allowable stress.

   2) This procedure applies only for Loading Cases 1, 2, and 3.

   3) This procedure applies to Configuration a when $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$.

   4) This procedure applies to Configurations b and c when $\sigma_s \le S_{PS,s}$.

   5) This procedure applies to Configurations e, f, and A when $\sigma_c \le S_{PS,c}$.

   6) This procedure may only be used once for each iteration of tubesheet, shell, and channel thickness and change of materials.

c) Calculation Procedure – After the calculation procedure given in paragraph 4.18.9.4 (STEP 1 through 10) has been performed for the elastic solution, an elastic-plastic calculation using the referenced steps from paragraph 4.18.8.4 shall be performed in accordance with the following procedure for each applicable loading case. Except for those quantities modified below, the quantities to be used for the elastic-plastic calculation shall be the same as those calculated for the corresponding elastic loading case.

   1) Define the maximum permissible bending stress limit in the shell and channel.

$$S_s^* = \min\left[S_{y,s}, \left(\frac{S_{PS,s}}{2}\right)\right] \qquad Configurations - a, b, c \tag{4.18.214}$$

$$S_c^* = \min\left[S_{y,c}, \left(\frac{S_{PS,c}}{2}\right)\right] \qquad Configurations - a, e, f \text{ and } A \tag{4.18.215}$$

2) Using bending stresses $\sigma_{s,b}$ and $\sigma_{c,b}$ calculated in STEP 10 for the elastic solution, determine $fact_s$ and $fact_c$ as follows.

$$fact_s = \min\left[\left(1.4 - \frac{0.4\left|\sigma_{s,b}\right|}{S_s*}\right), 1.0\right] \qquad Configurations - a,b,c \tag{4.18.216}$$

$$fact_c = \min\left[\left(1.4 - \frac{0.4\left|\sigma_{c,b}\right|}{S_c*}\right), 1.0\right] \qquad Configurations - a,e,f \text{ and } A \tag{4.18.217}$$

3) For Configuration a, if $fact_s = 1.0$ and $fact_c = 1.0$, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to paragraph 4.18.9.6.c.4. For Configurations b and c, if $fact_s = 1.0$, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to paragraph 4.18.8.6.c.4.

4) Calculate reduced values of E_s and E_c as follows:

$$E_s* = E_s \cdot fact_s \qquad Configurations - a,b,c \tag{4.18.218}$$

$$E_c* = E_c \cdot fact_c \qquad Configurations - a,e,f \text{ and } A \tag{4.18.219}$$

5) In STEP 2, recalculate k_s and λ_s by replacing E_s with E_s*, and k_c and λ_c by replacing E_c with E_c*.

6) In STEP 4, recalculate F, Φ, and Q_1.

7) In STEP 7, recalculate Q_2, Q_3, F_m, as applicable, and the tubesheet bending stress σ. If $\left|\sigma\right| \le 1.5S$, the design is acceptable and the calculation procedure is complete. Otherwise, the unit geometry shall be reconsidered.

4.18.9.6 Calculation Procedure for Effect of Radial Differential Thermal Expansion Adjacent to the Tubesheet

a) Scope

1) This procedure describes how to use the rules of paragraph 4.18.9.4 when the effect of radial differential thermal expansion between the tubesheet and integral shell or channel is to be considered.

2) This procedure shall be used when cyclic or dynamic reactions due to pressure or thermal variations are specified.

3) This procedure shall be used when specified by the user. The user shall provide the Manufacturer with the data necessary to determine the required tubesheet, channel, and shell metal temperatures.

4) Optionally, the designer may use this procedure to consider the effect of radial differential thermal expansion even when it is not required by paragraphs 4.18.9.5.a.2 or 4.18.9.5.a.3.

b) Conditions of Applicability – This calculation procedure applies only when the tubesheet is integral with the shell or channel (Configurations a, b, c, e, f and A).

c) Calculation Procedure – The calculation procedure outlined in paragraph 4.18.9.4 shall be performed only for Loading Cases 4, 5, 6 and 7, accounting for the following modifications.

1) Determine the average temperature of the unperforated rim T_r.

$$T_r = \frac{T' + T_s' + T_c'}{3} \qquad Configuration - a \tag{4.18.220}$$

$$T_r = \frac{T' + T_s'}{2} \qquad Configurations - b, c \tag{4.18.221}$$

$$T_r = \frac{T' + T_c'}{2} \qquad Configurations - e, f, A \tag{4.18.222}$$

For conservative values of P_s* and P_c*, $T_r = T'$ may be used.

2) Determine the average temperature of the shell T_s* and channel T_c* at their junction to the tubesheet using the equations shown below.

$$T_s* = \frac{T_s' + T_r}{2} \qquad Configurations - a, b, c \tag{4.18.223}$$

$$T_c* = \frac{T_c' + T_r}{2} \qquad Configurations - a, e, f, A \tag{4.18.224}$$

For conservative values of P_s* and P_c*, $T_s* = T_s'$ and $T_c* = T_c'$ may be used.

3) Calculate P_s^* and P_c^*.

$$P_s* = \frac{E_s t_s \left[\alpha_s' \left(T_s* - T_a \right) - \alpha' \left(T_r - T_a \right) \right]}{a_s} \qquad Configurations - a, b, c \tag{4.18.225}$$

$$P_s* = 0 \qquad Configurations - e, f, A \tag{4.18.226}$$

$$P_c* = \frac{E_c t_c \left[\alpha_c' \left(T_c* - T_a \right) - \alpha' \left(T_r - T_a \right) \right]}{a_c} \qquad Configurations - a, e, f, A \tag{4.18.227}$$

$$P_c* = 0 \qquad Configurations - b, c \tag{4.18.228}$$

4) In STEP 7, replace the equation for Q_2 with:

$$Q_2 = \frac{\left(\omega_s * P_s - \omega_c * P_t \right) - \left(\omega_s P_s * - \omega_c P_c * \right) + \dfrac{\gamma_b W}{2\pi}}{1 + \Phi Z_m} \tag{4.18.229}$$

5) In STEP 10, replace the equations for $\sigma_{s,b}$ and $\sigma_{c,b}$ with the following equations, where H is

given by Equation (4.18.122).

$$\sigma_{s,b}=\frac{6k_s}{t_s^2}\left[\beta_s\left(\delta_s P_s+\frac{a_s^2 P_s *}{E_s t_s}\right)+\frac{6\left(1-\left(\nu *\right)^2\right)}{E *}\left(\frac{a_o^3}{h^3}\right)\left(1+\frac{h\beta_s}{2}\right)H\right] \tag{4.18.230}$$

$$\sigma_{c,b}=\frac{6k_c}{t_c^2}\left[\beta_c\left(\delta_c P_t+\frac{a_c^2 P_c *}{E_c t_c}\right)-\frac{6\left(1-\left(\nu *\right)^2\right)}{E *}\left(\frac{a_o^3}{h^3}\right)\left(1+\frac{h\beta_c}{2}\right)H\right] \tag{4.18.231}$$

4.18.9.7 Calculation Procedure for Simply Supported Floating Tubesheets

4.18.9.7.1 Scope

This procedure describes how to use the rules of 4.18.9.4 when the effect of the stiffness of the integral channel and/or shell is not considered.

4.18.9.7.2 Conditions of Applicability

This calculation applies only when the tubesheet is integral with the shell or channel (Configurations a,b, c, e, f,and A.)

4.18.9.7.3 Calculation Procedure

The calculation procedure outlined in 4.18.9.4 shall be performed accounting for the following modifications;

a) Perform STEPs 1 through 9.

b) Perform STEP 10 except as follows:

   1) The shell (Configuration a,b,and c) is not required to meet a minimum length requirement.

   2) The channel (Configurations a, e, f,and A) is not required to meet a minimum length requirement.

   3) Acceptance Criteria

      i) Configuration a: If $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$, then the shell and channel are acceptable. Otherwise increase the thickness of the overstressed component(s) (shell and/or channel) and return to STEP 1.

      ii) Configuration b and c : If $\sigma_s \le S_{PS,s}$ then the shell is acceptable. Otherwise increase the thickness of the shell and return to STEP 1.

      iii) Configuration e, f and A: If $\sigma_c \le S_{PS,c}$, then the channel is acceptable. Otherwise increase the thickness of the channel and return to STEP 1.

c) Do not perform STEP 11

d) Repeat STEPs 1 - 7 for loading cases 1 - 3, with the following changes to STEP 2 until the tubesheet stress criteria have been met:

   1) Configurations a, b, and c: $\beta_s=0,\ k_s=0,\ \lambda_s=0,\ \delta_s=0$.

   2) Configurations a, e, f, and A: $\beta_c=0,\ k_c=0,\ \lambda_c=0,\ \delta_c=0$.

4.18.10 Tube-to-Tubesheet Welds

4.18.10.1 Scope

These rules provide a basis for establishing weld sizes and allowable joint loads for full strength and partial strength tube-to-tubesheet welds.

4.18.10.2 Definitions

a) Full Strength Weld – A full strength tube-to-tubesheet weld is one in which the design strength is equal to or greater than the axial tube strength, F_t. When the weld in a tube-to-tubesheet joint meets the requirements of paragraph 4.18.10.3, it is a full strength weld and the joint does not require qualification by shear load testing. Such a weld also provides tube joint leak tightness.

b) Partial Strength Weld – A partial strength weld is one in which the design strength is based on the mechanical and thermal axial tube loads (in either direction) that are determined from the actual design conditions. The maximum allowable axial load of this weld may be determined in accordance paragraph 4.18.10.4 or Annex 4.C. When the weld in a tube-to-tubesheet joint meets the requirements of paragraph 4.18.10.4, it is a partial strength weld and the joint does not require qualification by shear load testing. Such a weld also provides tube joint leak tightness.

c) Seal Weld – A tube-to-tubesheet seal weld is one used to supplement an expanded tube joint to ensure leak tightness. Its size has not been determined based on axial tube loading.

4.18.10.3 Full Strength Welds

Full strength welds shown in Figure 4.18.13 shall conform to the following requirements.

a) The size of a full strength weld shall be determined in accordance with paragraph 4.18.10.5.

b) The maximum allowable axial load in either direction on a tube-to-tubesheet joint with a full strength weld shall be determined as follows.

   1) For loads due to pressure-induced axial forces, $L_{max} = F_t$.

   2) For loads due to thermally-induced or pressure plus thermally-induced axial forces:

      i) $L_{max} = F_t$ for welded tube-to-tubesheet joints where the thickness through the weld throat is less than the nominal tube wall thickness t.

      ii) $L_{max} = 2F_t$ for welded tube-to-tubesheet joints where the thickness through the weld throat is greater than or equal to the nominal tube wall thickness t.

4.18.10.4 Partial Strength Welds

Partial strength welds shown in Figure 4.18.13 shall conform to the following requirements.

a) The size of a partial strength weld shall be determined in accordance with paragraph 4.18.10.5.

b) The maximum allowable axial load in either direction on a tube-to-tubesheet joint with a partial strength weld shall be determined as follows.

   1) For loads due to pressure-induced axial forces, $L_{max} = F_f + F_g$, but not greater than F_t.

   2) For loads due to thermally-induced or pressure plus thermally-induced axial forces:

      i) $L_{max} = F_f + F_g$ but not greater than F_t, for welded tube-to-tubesheet joints where the thickness through the weld throat is less than the nominal tube wall thickness t.

ii) $L_{\max} = 2\left(F_f + F_g\right)$ but not greater than $2F_t$, for welded tube-to-tubesheet joints where the thickness through the weld throat is greater than or equal to the nominal tube wall thickness t.

4.18.10.5 Weld Size Design Equations

a) The size of tube-to-tubesheet strength welds shown in Figure 4.18.13 shall conform to the following requirements

1) For fillet welds shown in Figure 4.18.13 Sketch (a):

i) Calculate the minimum required length of the fillet weld leg.

$$a_r = \sqrt{\left(0.75d_o\right)^2 + 2.73t\left(d_o - t\right)f_w f_d} - 0.75d_o \quad (4.18.232)$$

ii) For full strength welds, $a_f \geq \max\left[a_r, t\right]$.

iii) For partial strength welds, $a_f \geq a_r$.

2) For groove welds shown in Figure 4.18.13 Sketch (b):

i) Calculate the minimum required length of the groove weld leg.

$$a_r = \sqrt{\left(0.75d_o\right)^2 + 1.76t\left(d_o - t\right)f_w f_d} - 0.75d_o \quad (4.18.233)$$

ii) For full strength welds, $a_g \geq \max\left[a_r, t\right]$.

iii) For partial strength welds, $a_g \geq a_r$.

3) For combined groove and fillet welds shown in Figure 4.18.13 Sketch (c) where a_f is equal to a_g:

i) Calculate the minimum required length of the combined weld legs.

$$a_r = 2\left[\sqrt{\left(0.75d_o\right)^2 + 1.07t\left(d_o - t\right)f_w f_d} - 0.75d_o\right] \quad (4.18.234)$$

ii) For full strength welds, $a_c \geq \max\left[a_r, t\right]$.

iii) For partial strength welds, $a_c \geq a_r$.

iv) Calculate a_f and a_g using the following equations.

$$a_f = \frac{a_c}{2} \quad (4.18.235)$$

$$a_g = \frac{a_c}{2} \quad (4.18.236)$$

4) For combined groove and fillet welds shown in Figure 4.18.13 Sketch (d) where a_f is not equal to a_g, a_r shall be determined as follows:

i) Choose a_g and calculate the minimum required length of the fillet weld leg.

$$a_r = \sqrt{\left(0.75d_o\right)^2 + 2.73t\left(d_o - t\right) f_w f_d f_f} - 0.75d_o \tag{4.18.237}$$

ii) For full strength welds $a_c \ge \max\left[\left(a_r + a_g\right),\, t\right]$.

iii) For partial strength welds $a_c \ge \left(a_r + a_g\right)$.

iv) Calculate a_f using the following equation.

$$a_f = a_c - a_g \tag{4.18.238}$$

b) In weld strength factors used in subparagraph a) above shall be calculated using the following equations.

$$f_d = 1.0 \qquad \textit{for full strength welds} \tag{4.18.239}$$

$$f_d = \frac{F_d}{F_t} \qquad \textit{for partial strength welds} \tag{4.18.240}$$

$$f_f = 1 - \frac{F_g}{f_d F_t} \tag{4.18.241}$$

$$f_w = \frac{S_a}{S_w} \tag{4.18.242}$$

where

$$F_f = 0.55\pi a_f \left(d_o + 0.67a_f\right) S_w \tag{4.18.243}$$

$$F_g = 0.85\pi a_g \left(d_o + 0.67a_g\right) S_w \tag{4.18.244}$$

$$F_t = \pi t\left(d_o - t\right) S_a \tag{4.18.245}$$

4.18.11 Bellows Expansion Joints

Bellows expansion joints shall be designed in accordance with paragraph 4.19.

4.18.12 Flanged-and-Flued or Flanged-Only Expansion Joints

4.18.12.1 Design

Flanged-and-Flued or Flanged-Only expansion joints shall be designed in accordance with Part 5. The design considerations (i.e. load case combinations) in paragraph 4.18.8.3 shall be considered in the analysis.

4.18.12.2 Fabrication

The following requirements shall be met in the fabrication of expansion joint flexible elements.

a) All welded joints shall comply with the requirements of Part 6.

b) All longitudinal and circumferential weld seams shall be butt-type full penetration welds; Type 1 welds in accordance with paragraph 4.2.

c) Longitudinal welds shall be ground flush and smooth on both inside and outside surfaces prior to being formed into expansion elements.

d) Other than the shell attachment welds and flange welds, no circumferential welds are permitted in the fabrication of flexible elements, i.e., inner torus, annular plate, outer torus, unless the welds are ground flush and fully radiographed.

e) Flexible elements shall be attached by full penetration circumferential welds.

f) Nozzles, backing strips, clips or other attachments shall not be located in highly stressed areas of the expansion joint, i.e. inner torus, annular plate, and outer torus.

4.18.12.3 Examination

The following examinations are required to verify the integrity of expansion joints.

a) All expansion joint flexible elements shall be visually examined and found to be free of unacceptable imperfections, such as notches, crevices, weld spatter, etc., which may serve as points of local stress concentration. Suspect surface areas shall be further examined by liquid penetrant or magnetic particle examination.

b) Longitudinal welds shall be fully radiographed in accordance with Part 7. All full penetration butt-type welds shall be examined 100% on both sides by liquid penetrant or magnetic particle examination after forming.

c) The circumferential welds within the expansion joint and attaching the expansion joint to the shell shall be examined 100% on both sides, where accessible, by liquid penetrant or magnetic particle examination. The accessibility of welds shall be subject to the acceptance of the Inspector.

4.18.12.4 Pressure Test Requirements

The pressure testing requirements for expansion joints shall be as follows.

a) The completed expansion joint shall be subject to a pressure test in accordance with paragraph 4.1.6.2 and Part 8. The pressure testing of an expansion joint may be performed as part of the final vessel pressure test, provided the joint is accessible for inspection during pressure testing.

b) Expansion joint restraining elements shall also be pressure tested in accordance with paragraph 4.1.6.2 and Part 8 as a part of the initial expansion joint pressure test or as part of the final vessel pressure test after installation of the joint.

c) In addition to inspecting the expansion joint for leaks during the pressure test, expansion joints shall be inspected before, during, and after the pressure test for visible permanent distortion.

4.18.12.5 Marking and Reports

a) The expansion joint Manufacturer, whether the vessel Manufacturer or a parts Manufacturer, shall have a valid ASME Code U2 Certificate of Authorization and shall complete the appropriate Data Report in accordance with Part 2.

b) The Manufacturer responsible for the expansion joint design shall include the following additional data and statements on the appropriate Data Report.

   1) Axial movement (+ and -), associated design life in cycles, and associated loading condition, if applicable;

   2) Spring rate; and

   3) The expansion joint has been constructed to the rules of this paragraph.

c) A parts Manufacturer shall identify the vessel for which the expansion joint is intended on the Partial Data Report.

d) Markings shall not be stamped on the flexible elements of the expansion joint.

4.18.13 Pressure Test Requirements

The shell side and the tube side of the heat exchanger shall be subjected to a pressure test in accordance with paragraph 4.1 and Part 8.

4.18.14 Heat Exchanger Marking and Reports

4.18.14.1 Required Marking

The marking of heat exchangers shall be in accordance with Annex 2.F using the specific requirements for combination units (multi-chamber vessels). When the markings are grouped in one location and abbreviations for each chamber are used, they shall be as follows:

a) The chambers shall be abbreviated SHELL for shell side and TUBES for tube side. This abbreviation shall precede the appropriate design data. For example, use the following for the shell side maximum allowable working pressure and for the tube side maximum allowable working pressure:

   1) SHELL FV&2000 kPa (FV&300 psi) at 280°C (500°F)

   2) TUBES 1000 kPa (150 psi) at 175°C (350°F)

b) When the markings are different for each chamber, the chambers shall be abbreviated with a S for shell side and T for the tube side. For example, use “F-T” for forged construction on the tube side.

4.18.14.2 Supplemental Marking

A supplemental tag or marking shall be supplied on the heat exchanger to caution the user if there are any restrictions on the design, testing, or operation of the heat exchanger. Supplemental marking shall be required for, but not limited to, the following:

a) Common Elements – Shell-and-tube heat exchangers are combination units as defined in paragraph 4.1.8.1, and the tubes and tubesheets are common elements. The following marking is required when the common elements are designed for conditions less severe than the design conditions for which its adjacent chambers are stamped.

   1) Differential Pressure Design – When common elements such as tubes and tubesheets are designed for a differential design pressure, the heat exchanger shall be marked “Differential Design” in addition to meeting the requirements of paragraph 4.1.8.1. If the tubes and tubesheets are designed for a differential pressure of 150 psi, an example of the marking would be:

      DIFFERENTIAL DESIGN: TUBES & TUBESHEETS 150 psi

   2) Mean Metal Temperature Design – When common elements such as tubes and tubesheets are designed for a maximum mean metal design temperature that is less than the maximum of the shell side and tube side design temperatures, the heat exchanger shall be marked “Max Mean Metal Temp” in addition to meeting the requirements of paragraph 4.1.8.1. If the tubes are designed for a maximum mean metal temperature of 400˚F, an example of the marking would be:

      MAX MEAN METAL TEMP: TUBES 400˚F

b) Fixed Tubesheet Heat Exchangers – Fixed tubesheet heat exchangers shall be marked with a caution such as follows:

   “The Code required pressures and temperatures marked on the heat exchanger relate to the basic design conditions. The heat exchanger design has been evaluated for specific operating conditions and

shall be re-evaluated before it is operated at different operating conditions."

4.18.14.3 Manufacturer's Data Reports

When common elements such as tubes and tubesheets are designed for a differential pressure or a mean metal temperature or both, that is less severe than the design conditions for which its adjacent chambers are stamped, the data for each common element that differs from the data for the corresponding chamber shall be documented as required in paragraph 4.1.8.2 in the "Remarks" section of the Manufacturer's Data Report.

4.18.15 Nomenclature

a) Nomenclature for tubesheet flanged extension (paragraph 4.18.5).

G — diameter of gasket load reaction
- $= G_c$ for U-tube tubesheet Configuration b
- $= G_s$ for U-tube tubesheet Configuration e
- $= G_c$ for fixed tubesheet Configuration b
- $= G_c$ for stationary tubesheet Configuration b of a floating tubesheet exchanger
- $= G_s$ for stationary tubesheet Configuration e of a floating tubesheet exchanger
- $= G_c$ for floating tubesheet Configuration B of a floating tubesheet exchanger
- $= G_c$ or G_s for tubesheet Configuration d when applicable (eg. hydrotest)

h_g — gasket moment arm, equal to the radial distance from the center line of the bolts to the line of the gasket reaction (see paragraph 4.16).

h_r — minimum required thickness of the tubesheet flanged extension.

S — allowable stress from Annex 3.A for the material of the tubesheet extension at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply.

W — flange design bolt load.
- $= W_g$ for gasket seating conditions.
- $= W_o$ for operating conditions.

b) Nomenclature for determining tubesheet characteristics (see paragraph 4.18.6)

A_L — total area of untubed lanes, $A_L = U_{L1}L_{L1} + U_{L2}L_{L2} + ... + U_{Ln}L_{Ln}$ (limited to $4D_0 p$).

A_p — total area enclosed by C_p.

c_t — tubesheet corrosion allowance on the tube side, $c_t = 0$ for the uncorroded condition.

C_p — perimeter of the tube layout measured stepwise in increments of one tube pitch from the center-to-center of the outer most tubes (see Figure 4.18.14).

D_o — equivalent diameter of outer tube limit circle.

d — diameter of tube hole.

d_t — nominal outside diameter of tubes.

d^* — effective tube hole diameter.

E — modulus of elasticity for tubesheet material at tubesheet design temperature.

E_{tT} — modulus of elasticity for tube material at tubesheet design temperature, T.

E^* — effective modulus of elasticity of tubesheet in perforated region, T.

h — tubesheet thickness.

h_g tube side pass partition groove depth.

$h_g^{'}$ effective tube side pass partition groove depth.

$L_{L1}, L_{L2},...$ length(s) of untubed lane(s).

l_{tx} expanded length of tube in tubesheet, $0 \le l_{tx} \le h$. An expanded tube-to-tubesheet joint is produced by applying pressure inside the tube such that contact is established between the tube and tubesheet. In selecting an appropriate value of expanded length, the designer shall consider the degree of initial expansion, differences in thermal expansion, or other factors that could result in loosening of the tubes within the tubesheet.

p tube pitch.

$p*$ effective tube pitch.

r_o radius to outermost tube hole center.

S allowable stress from Annex 3.A for tubesheet material at tubesheet design temperature, T.

S_{tT} allowable stress from Annex 3.A for tube material at tubesheet design temperature, T. For a welded tube or pipe, use the allowable stress for the equivalent seamless product. If the allowable stress for the equivalent seamless product is not available, then divide the allowable stress of the welded product by 0.85.

T tubesheet design temperature.

t_t nominal tube wall thickness.

$U_{L1}, U_{L2},...$ center-to-center distance(s) between adjacent tube rows of untubed lane(s), but not to exceed $4p$.

μ basic ligament efficiency for shear.

μ^* effective ligament efficiency for bending.

ν^* effective Poisson's ratio in perforated region of tubesheet.

ρ tube expansion depth ratio.

c) Nomenclature for the design of U-tube tubesheets (see paragraphs 4.18.7)

A outside diameter of tubesheet

C bolt circle diameter (see paragraph 4.16)

D_c inside channel diameter

D_s inside shell diameter

E modulus of elasticity for tubesheet material at design temperature

E_c modulus of elasticity for channel material at design temperature

E_s modulus of elasticity for shell material at design temperature

G_1 midpoint of contact between flange and tubesheet

G_c diameter of channel gasket load reaction (see paragraph 4.16)

G_s diameter of shell gasket load reaction (see paragraph 4.16)

h tubesheet thickness

P_s shell side internal design pressure, for shell side vacuum use a negative value.

P_t tube side internal design pressure, for tube side vacuum use a negative value.

S allowable stress from Annex 3.A for tubesheet material at tubesheet design temperature

S_c allowable stress from Annex 3.A for channel material at design temperature. For a welded

tube or pipe, use the allowable stress for the equivalent seamless product. If the allowable stress for the equivalent seamless product is not available, then divide the allowable stress of the welded product by 0.85.

S_s allowable stress from Annex 3.A for shell material at design temperature. For a welded tube or pipe, use the allowable stress for the equivalent seamless product. If the allowable stress for the equivalent seamless product is not available, then divide the allowable stress of the welded product by 0.85.

$S_{y,c}$ yield strength from Annex 3.D for channel material at design temperature

$S_{y,s}$ yield strength from Annex 3.D for shell material at design temperature

$S_{PS,c}$ allowable primary plus secondary stress evaluated using paragraph 5.5.6.1.d for channel material at design temperature $S_{PS,s}$ allowable primary plus secondary stress evaluated using paragraph 5.5.6.1.d for shell material at design temperature

t_c channel thickness

t_s shell thickness

W_c channel flange design bolt load for the gasket seating condition (see paragraph 4.16)

W_s shell flange design bolt load for the gasket seating condition (see paragraph 4.16)

$W_{\max}$ maximum flange design bolt load.

ν_c Poisson's ratio of channel material

ν_s Poisson's ratio of shell material

d) Nomenclature for the design of fixed or floating tubesheets (see paragraphs 4.18.8 and for paragraph 4.18.9)

A outside diameter of tubesheet

a_c radial channel dimension

a_o equivalent radius of outer tube limit circle

a_s radial shell dimension

C bolt circle diameter (see paragraph 4.16)

d_t nominal outside diameter of tubes

D_c inside channel diameter

D_J inside diameter of the expansion joint at its convolution height

D_s inside shell diameter

E modulus of elasticity for tubesheet material at T

E_c modulus of elasticity for channel material at T_c

E_s modulus of elasticity for shell material at T_s

$E_{s,1}$ modulus of elasticity for shell material adjacent to the tubesheet at T_s

$E_{s,w}$ joint efficiency (longituidinal stress) for shell

E_t modulus of elasticity for tube material at T_t

G_1 midpoint of contact between flange and tubesheet

G_c diameter of channel gasket load reaction (see paragraph 4.16)

G_s diameter of shell gasket load reaction (see paragraph 4.16)

h tubesheet thickness

J ratio of expansion joint to shell axial rigidity ($J = 1.0$ if no expansion joint)

K_J axial rigidity of expansion joint, total force/elongation

k constant accounting for the method of support for the unsupported tube span under consideration
= 0.6 for unsupported spans between two tubesheets,
= 0.8 for unsupported spans between a tubesheet and a tube support,
= 1.0 for unsupported spans between two tube supports.

l unsupported tube span under consideration

l_1, l_1' lengths of shell thickness $t_{s,1}$ adjacent to the tubesheets

L tube length between inner tubesheet faces, $L = L_t - 2h$

L_t tube length between outer tubesheet faces

N_t number of tubes

P_e effective pressure acting on tubesheet

P_s shell side internal design pressure, for shell side vacuum use a negative value.

P_t tube side internal design pressure, for tube side vacuum use a negative value.

S allowable stress from Annex 3.A for tubesheet material tubesheet material at T

S_c allowable stress from Annex 3.A for channel material at T_c. For a welded tube or pipe, use the allowable stress for the equivalent seamless product. If the allowable stress for the equivalent seamless product is not available, then divide the allowable stress of the welded product by 0.85.

S_s allowable stress from Annex 3.A for shell material at T_s. For a welded tube or pipe, use the allowable stress for the equivalent seamless product. If the allowable stress for the equivalent seamless product is not available, then divide the allowable stress of the welded product by 0.85.

$S_{s,1}$ allowable stress from Annex 3.A for shell material adjacent to the tubesheets at T_s

$S_{s,b}$ maximum allowable longituidinal stress in accordance with 4.4.12.2 for the shell

$S_{s,b,1}$ maximum allowable longituidinal stress in accordance with 4.4.12.2 for the shell adjacent to the tubesheets.

S_t allowable stress from Annex 3.A for tube material at T_t. For a welded pipe or tube, use the allowable stress from Annex 3.A for the equivalent seamless product. When the allowable stress for the equivalent seamless product is not available, divide the allowable stress of the welded product by 0.85.

S_y yield strength from Annex 3.D for tubesheet material at T

$S_{y,c}$ yield strength from Annex 3.D for channel material at T_c

$S_{y,s}$ yield strength from Annex 3.D for shell material at T_s

$S_{y,s,1}$ yield strength from Annex 3.D for shell material adjacent to the tubesheets at T_s

$S_{y,t}$ yield strength from Annex 3.D for tube material at T_t

S_{PS} allowable primary plus secondary stress evaluated using paragraph 5.5.6.1.d for tubesheet material at temperature T

$S_{PS,c}$ allowable primary plus secondary stress evaluated using paragraph 5.5.6.1.d for channel material at temperature T_c

$S_{PS,s}$ allowable primary plus secondary stress evaluated using paragraph 5.5.6.1.dfor shell material at temperature T_s

$S_{PS,s,1}$ allowable primary plus secondary stress evaluated using paragraph 5.5.6.1.d for shell material adjacent to the tubesheet at temperature T_s

T tubesheet design temperature

T_a ambient temperature, 20°C (70°F)

T_c channel design temperature

T_s shell design temperature

T_t tube design temperature

$T_{s,m}$ mean shell metal temperature along shell length

$T_{t,m}$ mean tube metal temperature along tube length

T' tubesheet metal temperature at the rim

T_c' channel metal temperature at the tubesheet

T_s' shell metal temperature at the tubesheet

t_c channel thickness

t_s shell thickness

$t_{s,1}$ shell thickness adjacent to the tubesheets

t_t nominal tube wall thickness

W flange design bolt load for the gasket seating condition (see paragraph 4.16)

W_t tube-to-tubesheet joint load

$\alpha_{s,m}$ mean coefficient of thermal expansion of shell material at $T_{s,m}$

$\alpha_{s,m,1}$ mean coefficient of thermal expansion of shell material adjacent to the tubesheets at $T_{s,m}$

$\alpha_{t,m}$ mean coefficient of thermal expansion of tube material at $T_{t,m}$

α' mean coefficient of thermal expansion of tubesheet material at T'

α_c' mean coefficient of thermal expansion of channel material at T_c'

α_s' mean coefficient of thermal expansion of shell material at T_s'

γ axial differential thermal expansion between tubes and shell

ν Poisson's ratio of tubesheet material

ν_c Poisson's ratio of channel material

ν_s Poisson's ratio of shell material

ν_t Poisson's ratio of tube material

e) Nomenclature for tube-to-tubesheet welds (see paragraph 4.18.10)

a_c length of the combined weld legs measured parallel to the longitudinal axis of the tube at its outside diameter

a_f fillet weld leg

a_g groove weld leg

a_r minimum required length of the weld leg(s) under consideration

d_o nominal tube outside diameter

f_d ratio of the design strength to the tube strength

f_f ratio of the fillet weld strength to the design strength

f_w weld strength factor

F_d design strength, but not greater than F_t

F_f fillet weld strength, but not greater than F_t

F_g groove weld strength, but not greater than F_t

F_t axial tube strength

$L_{\max}$ maximum allowable axial load in either direction on the tube-to-tubesheet joint

t nominal tube wall thickness

S_a allowable stress from Annex 3.A of the tube

S_t allowable stress from Annex 3.A of the material to which the tube is welded

S_w allowable stress in weld, $S_w = \min\left[S_a, S_t\right]$

4.18.16 Tables

Table 4.18.1 – Effective Elastic Modulus And Poisson's Ratio For A Perforated Plate With An Equilateral Triangular Hole Pattern

$\frac{h}{p}$	A_0	A_1	A_2	A_3	A_4
0.10	0.0353	1.2502	-0.0491	0.3604	-0.6100
0.25	0.0135	0.9910	1.0080	-1.0498	0.0184
0.50	0.0054	0.5279	3.0461	-4.3657	1.9435
2.00	-0.0029	0.2126	3.9906	-6.1730	3.4307
$\frac{h}{p}$	B_0	B_1	B_2	B_3	B_4
0.10	-0.0958	0.6209	-0.8683	2.1099	-1.6831
0.15	0.8897	-9.0855	36.1435	-59.5425	35.8223
0.25	0.7439	-4.4989	12.5779	-14.2092	5.7822
0.50	0.9100	-4.8901	12.4325	-12.7039	4.4298
1.00	0.9923	-4.8759	12.3572	-13.7214	5.7629
2.00	0.9966	-4.1978	9.0478	-7.9955	2.2398

Notes:

1. $E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$
2. $\nu^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$
3. These coefficients are only valid for $0.1 \le \mu^* \le 0.6$. Data for the range $0.1 \le \mu^* \le 1.0$ is provided in Part 5, Annex 5.E.
4. If $h/p < 0.1$, use $h/p = 0.1$.
5. If $h/p > 2.0$, use $h/p = 2.0$.

Table 4.18.2 – Effective Elastic Modulus And Poisson's Ratio For A Perforated Plate With A Square Hole Pattern

$\frac{h}{p}$	A_0	A_1	A_2	A_3	A_4
0.10	0.0676	1.5756	-1.2119	1.7715	-1.2628
0.25	0.0250	1.9251	-3.5230	6.9830	-5.0017
0.50	0.0394	1.3024	-1.1041	2.8714	-2.3994
2.00	0.0372	1.0314	-0.6402	2.6201	-2.1929
$\frac{h}{p}$	B_0	B_1	B_2	B_3	B_4
0.10	-0.0791	0.6008	-0.3468	0.4858	-0.3606
0.15	0.3345	-2.8420	10.9709	-15.8994	8.3516
0.25	0.4296	-2.6350	8.6864	-11.5227	5.8544
0.50	0.3636	-0.8057	2.0463	-2.2902	1.1862
1.00	0.3527	-0.2842	0.4354	-0.0901	-0.1590
2.00	0.3341	0.1260	-0.6920	0.6877	-0.0600

Notes:

1. $E^*/E = A_0 + A_1\mu^* + A_2(\mu^*)^2 + A_3(\mu^*)^3 + A_4(\mu^*)^4$
2. $\nu^* = B_0 + B_1\mu^* + B_2(\mu^*)^2 + B_3(\mu^*)^3 + B_4(\mu^*)^4$
3. These coefficients are only valid for $0.1 \le \mu^* \le 0.6$. Data for the range $0.1 \le \mu^* \le 1.0$ is provided in Part 5, Annex 5.E.
4. If $h/p < 0.1$, use $h/p = 0.1$.
5. If $h/p > 2.0$, use $h/p = 2.0$.

Table 4.18.3 – Evaluation Of Z_a, Z_d, Z_v, Z_w, Z_m, And F_m

<table>
<tr><th>Evaluation Of Kelvin Functions ber, bei, ber' and bei' relative to x</th></tr>
<tr><td>

$$ber(x)=\sum_{n=0}^{n=m-1}(-1)^n\frac{(x/2)^{4n}}{(2n)!^2}=1-\frac{(x/2)^4}{(2)!^2}+\frac{(x/2)^8}{(4)!^2}-\frac{(x/2)^{12}}{(6)!^2}+\ldots$$

$$bei(x)=\sum_{n=1}^{n=m}(-1)^{n-1}\frac{(x/2)^{4n-2}}{(2n-1)!^2}=\frac{(x/2)^2}{(1)!^2}-\frac{(x/2)^6}{(3)!^2}+\frac{(x/2)^{10}}{(5)!^2}-\ldots$$

$$ber'(x)=\sum_{n=1}^{n=m}(-1)^n\frac{2n(x/2)^{4n-1}}{(2n)!^2}=-\frac{2(x/2)^3}{(2)!^2}+\frac{4(x/2)^7}{(4)!^2}-\frac{6(x/2)^{11}}{(6)!^2}+\ldots$$

$$bei'(x)=\sum_{n=1}^{n=m}(-1)^{n-1}\frac{(2n-1)(x/2)^{4n-3}}{(2n-1)!^2}=\frac{(x/2)^1}{(1)!^2}-\frac{3(x/2)^5}{(3)!^2}+\frac{5(x/2)^9}{(5)!^2}-\ldots$$

Note: Use $m=4+X_a/2$ terms (rounded to the nearest integer) to obtain an adequate approximation of the Kelvin Functions and their derivatives.

</td></tr>
<tr><th>ψ_i Functions For Determination of Z_a, Z_d, Z_v, Z_w, Z_m, and F_m relative to x</th></tr>
<tr><td>

$$\psi_1(x)=bei(x)+\left(\frac{1-\nu^*}{x}\right)ber'(x)$$

$$\psi_2(x)=ber(x)-\left(\frac{1-\nu^*}{x}\right)bei'(x)$$

</td></tr>
<tr><th>Evaluation Of Z_a, Z_d, Z_v, Z_w, Z_m at X_a</th></tr>
<tr><td>

$$Z_d=\frac{ber(X_a)\psi_2(X_a)+bei(X_a)\psi_1(X_a)}{X_a^3 Z_a}$$

$$Z_v=\frac{ber'(X_a)\psi_2(X_a)+bei'(X_a)\psi_1(X_a)}{X_a^2 Z_a}$$

$$Z_w=\frac{ber'(X_a)ber(X_a)+bei'(X_a)bei(X_a)}{X_a^2 Z_a}$$

$$Z_m=\frac{\left[ber'(X_a)\right]^2+\left[bei'(X_a)\right]^2}{X_a Z_a}$$

where

$$Z_a=bei'(X_a)\psi_2(X_a)-ber'(X_a)\psi_1(X_a)$$

</td></tr>
<tr><th>Evaluation Of F_m from $0\rightarrow X_a$</th></tr>
<tr><td>

Calculate the functions $Q_m(x)$ and $Q_v(x)$ relative to x:

</td></tr>
</table>

Table 4.18.3 – Evaluation Of Z_a, Z_d, Z_v, Z_w, Z_m, And F_m

$$Q_m(x) = \frac{bei'(X_a)\psi_2(x) - ber'(X_a)\psi_1(x)}{Z_a}$$

$$Q_v(x) = \frac{\psi_1(X_a)\psi_2(x) - \psi_2(X_a)\psi_1(x)}{X_a Z_a}$$

For each loading case (note that Q_3 is a dependent on the load case being evaluated), calculate $F_m(x)$ relative to x:

$$F_m(x) = \frac{Q_v(x) + Q_3 \cdot Q_m(x)}{2}$$

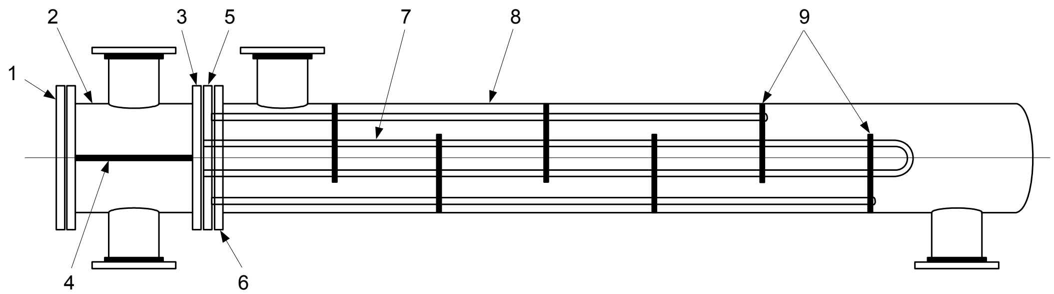
F_m is the maximum of the absolute value of $F_m(x)$ as x varies from $0 \rightarrow X_a$ such that $0 \le x \le X_a$

$$F_m = \max\left[\left|F_m(x)\right|\right]$$

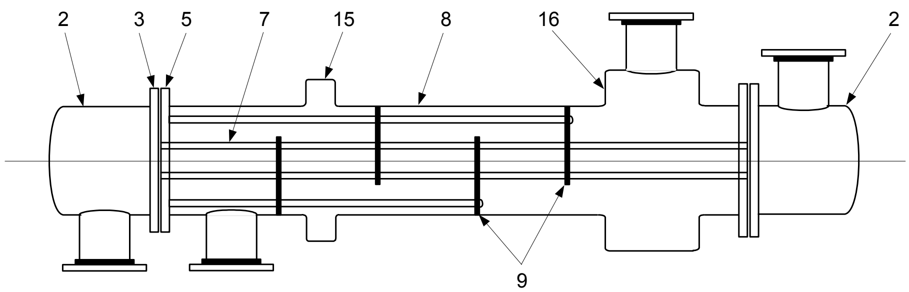
Table 4.18.4 – Evaluation Of $F_{t,\min}$ and $F_{t,\max}$

Formulas For The Determination of $F_{t,\min}$ and $F_{t,\max}$
Calculate the Kelvin functions, the ψ_i functions and Z_a from Table 4.18.3 Caclulate functions $Z_d(x)$ and $Z_w(x)$ relative to x: $$Z_d(x)=\frac{\psi_2(X_a)\cdot ber(x)+\psi_1(X_a)\cdot bei(x)}{X_a^3\cdot Z_a}$$ $$Z_w(x)=\frac{ber'(X_a)\cdot ber(x)+bei'(X_a)\cdot bei(x)}{X_a^2\cdot Z_a}$$ For each loading case, calculate $F_t(x)$ relative to x: When $P_e \neq 0$ $$F_t(x)=\left[Z_d(x)+Q_3\cdot Z_w(x)\right]\cdot\frac{X_a^4}{2}$$ When $P_e = 0$ $$F_t(x)=Z_w(x)\cdot\frac{X_a^4}{2}$$ Calculate the minimum and maximum values, $F_{t,\min}$ and $F_{t,\max}$, of $F_t(x)$ as x varies $0\rightarrow X_a$ such that $0\le x\le X_a$. $F_{t,\min}$ and $F_{t,\max}$ may be positive or negative. $F_{t,\min}=\min\left[F_t(x)\right]$ $F_{t,\max}=\max\left[F_t(x)\right]$

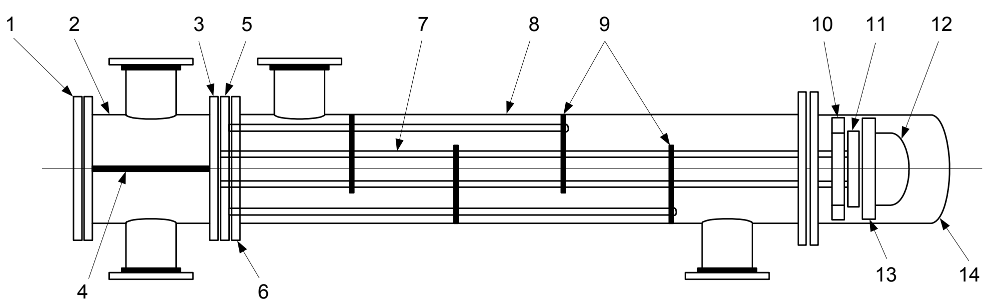
4.18.17 Figures



(a) U-Tube Heat Exchanger



(b) Fixed Tubesheet Heat Exchanger

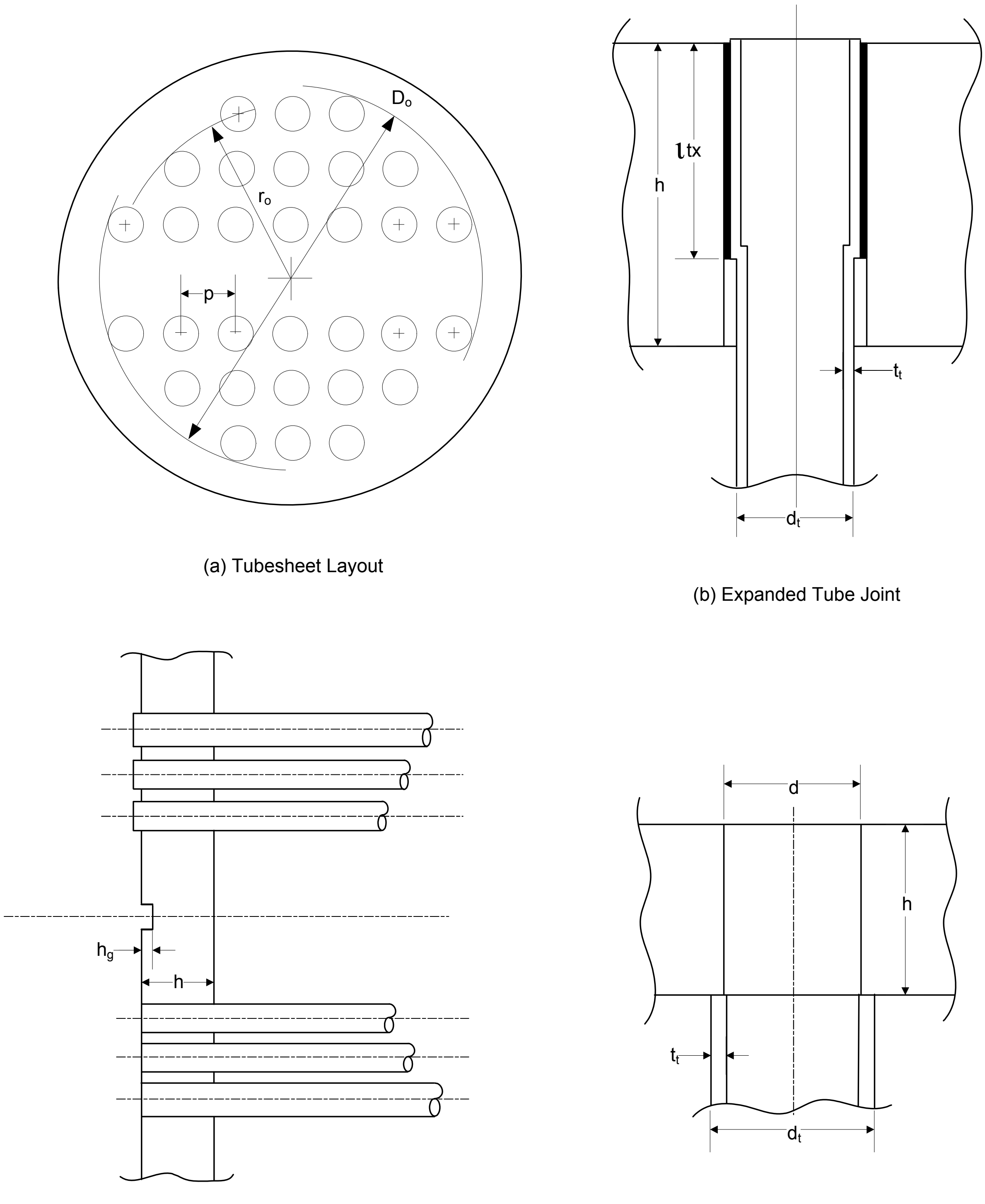


(c) Floating Tubesheet Heat Exchanger

Parts:

1 Channel Cover (Bolted Flat Cover)
2 Channel
3 Channel Flange
4 Pass Partition
5 Stationary Tubesheet
6 Shell Flange
7 Tubes
8 Shell
9 Baffles or Support Plates
10 Floating Head Backing Device
11 Floating Tubesheet
12 Floating Head
13 Floating Head Flange
14 Shell Cover
15 Expansion Joint
16 Distribution or Vapor Belt

Figure 4.18.1 – Terminology of Heat Exchanger Components



(a) Tubesheet Layout

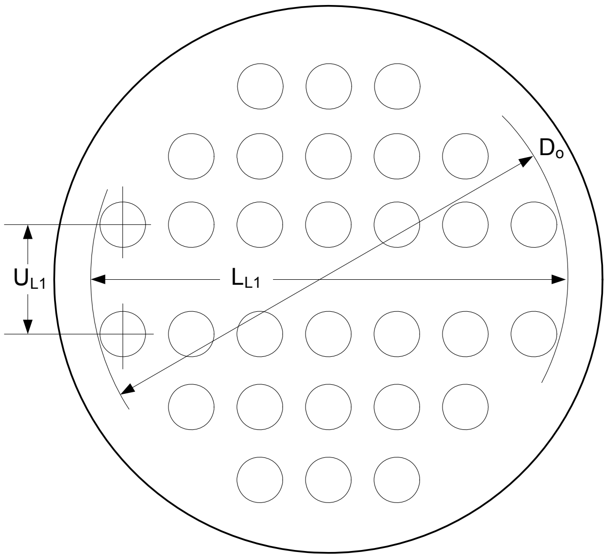
(b) Expanded Tube Joint

(c) Tube Side Pass Partition Groove Depth

(d) Tubes Welded To Backside Of Tubesheet

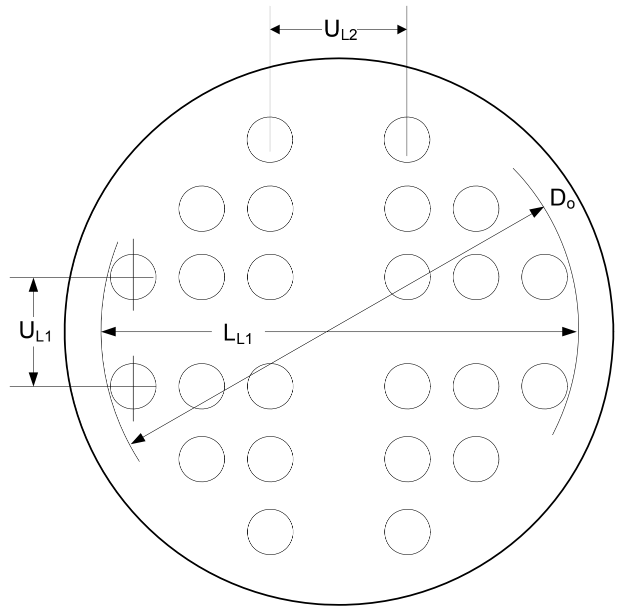
Note: $d_t - 2t_t \leq d < d_t$

Figure 4.18.2 – Tubesheet Geometry



$A_L = U_{L1}L_{L1}$

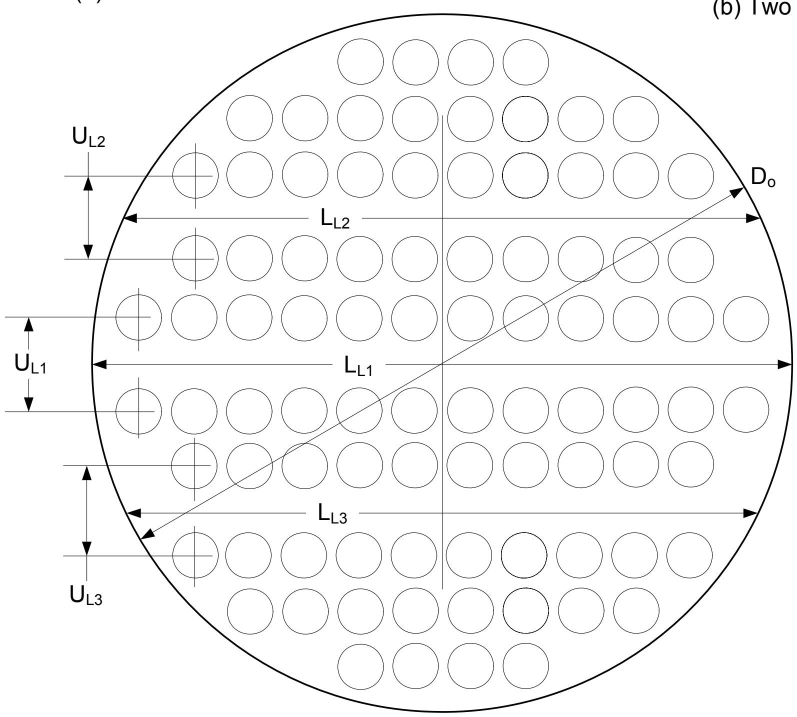
(a) One Lane



$L_{L2} = L_{L1} - U_{L1}$

$A_L = U_{L1}L_{L1} + U_{L2}L_{L2}$

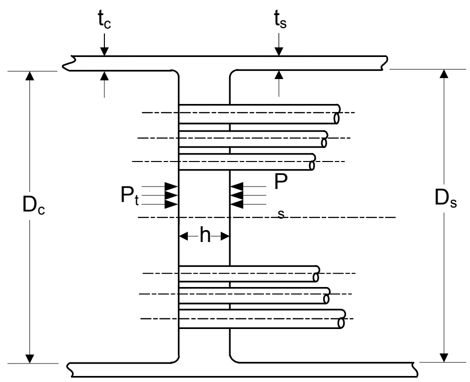
(b) Two Lanes



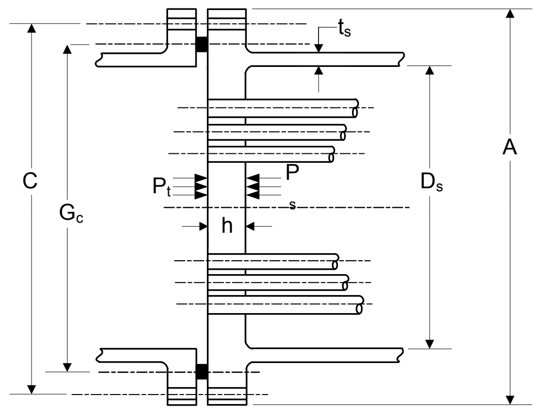
$A_L = U_{L1}L_{L1} + U_{L2}L_{L2} + U_{L3}L_{L3}$

(c) Three Lanes

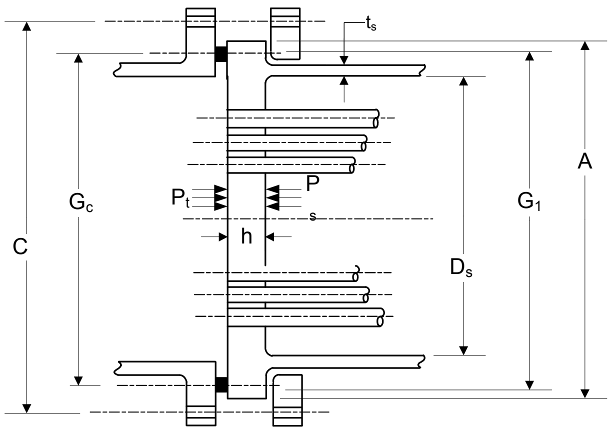
Figure 4.18.3 – Typical Untubed Lane Configurations



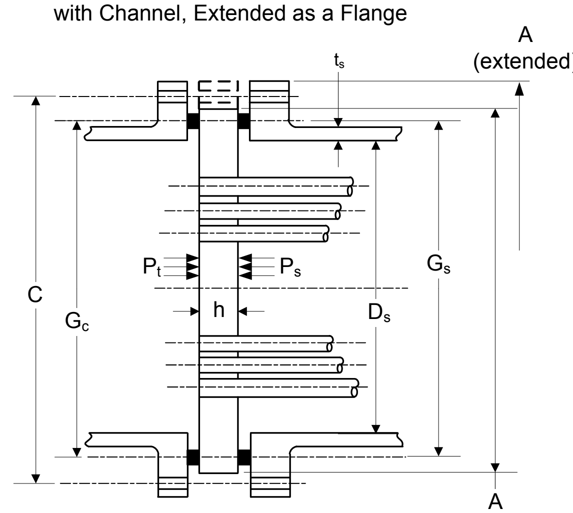
(a) Configuration a:
Tubesheet Integral with Shell and Channel



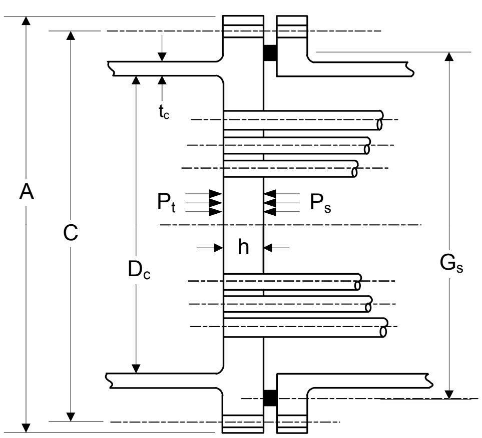
(b) Configuration b:
Tubesheet Integral with Shell and Gasketed with Channel, Extended as a Flange



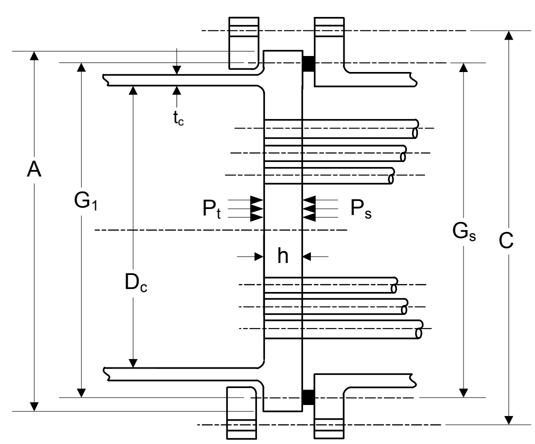
(c) Configuration c:
Tubesheet Integral with Shell and Gasketed with Channel, Not Extended as a Flange



(d) Configuration d:
Tubesheet Integral with Shell and Channel

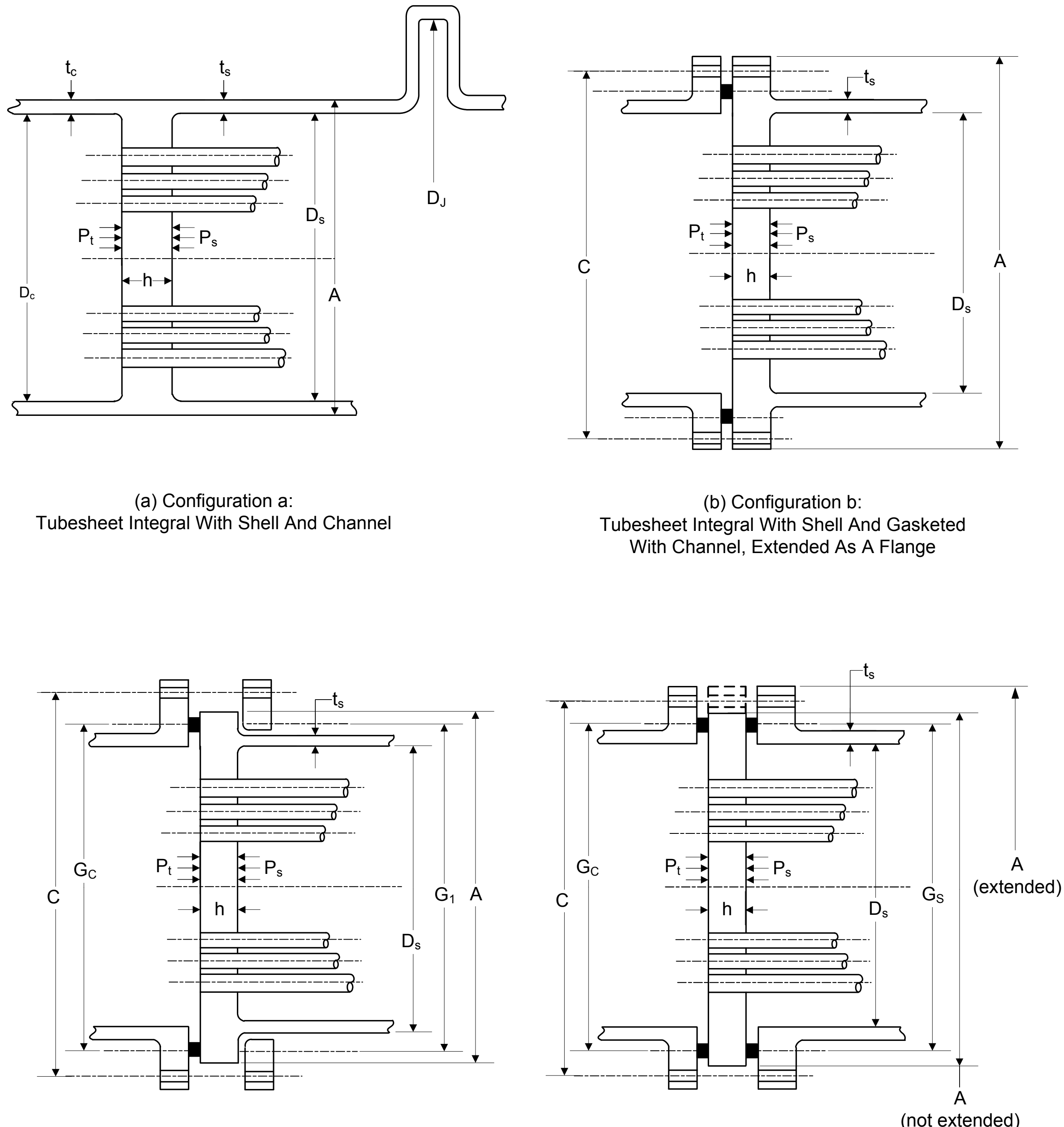


(e) Configuration e:
Tubesheet Gasketed with Shell and Integral with Channel, Extended as a Flange



(f) Configuration f:
Tubesheet Gasketed with Shell and Integral with Channel, Not Extended as a Flange

Figure 4.18.4 – U-Tube Tubesheet Configurations



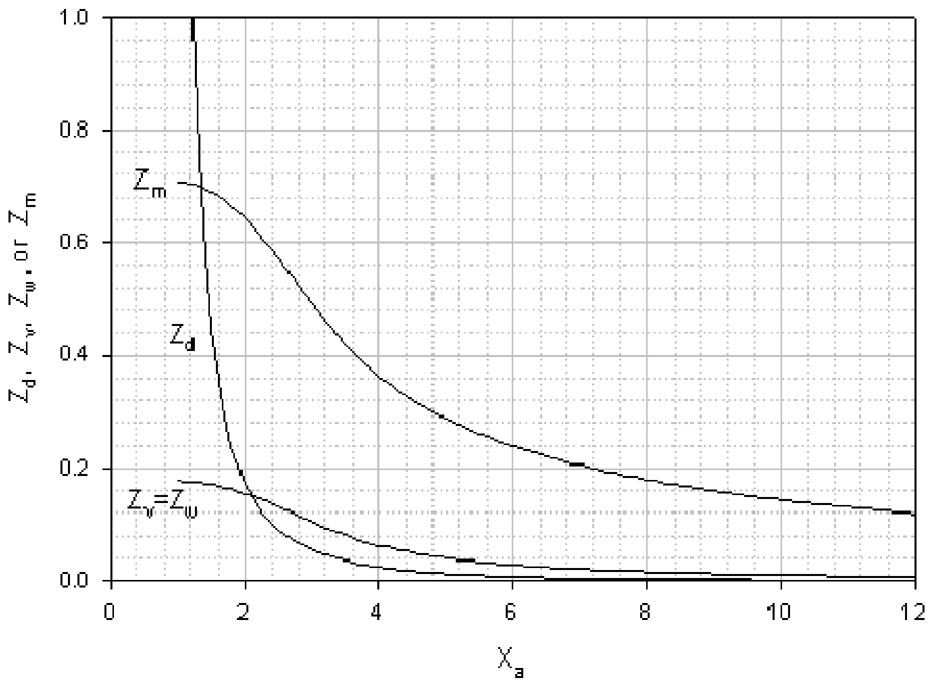
(a) Configuration a:
Tubesheet Integral With Shell And Channel

(b) Configuration b:
Tubesheet Integral With Shell And Gasketed With Channel, Extended As A Flange

(c) Configuration c:
Tubesheet Integral With Shell And Gasketed With Channel, Not Extended As A Flange

(d) Configuration d:
Tubesheet Gasketed With Shell And Channel

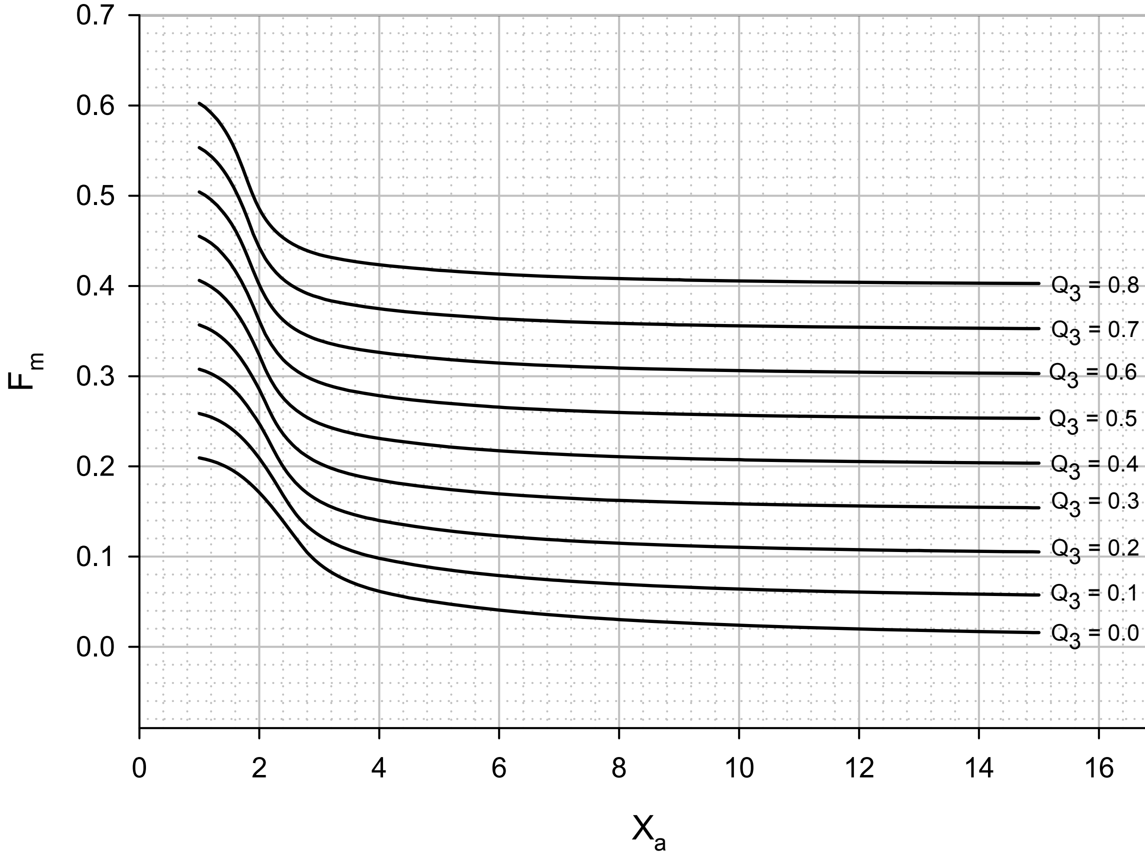
Figure 4.18.5 – Fixed Tubesheet Configurations



Notes:

1. The curves for Z_d, Z_v, Z_w, and Z_m are valid for $\nu^* = 0.4$. These curves are sufficiently accurate for other values of ν^*.
2. If $X_a > 12$, Z_d, Z_v, Z_w, and Z_m shall be calculated using Table 4.18.3.

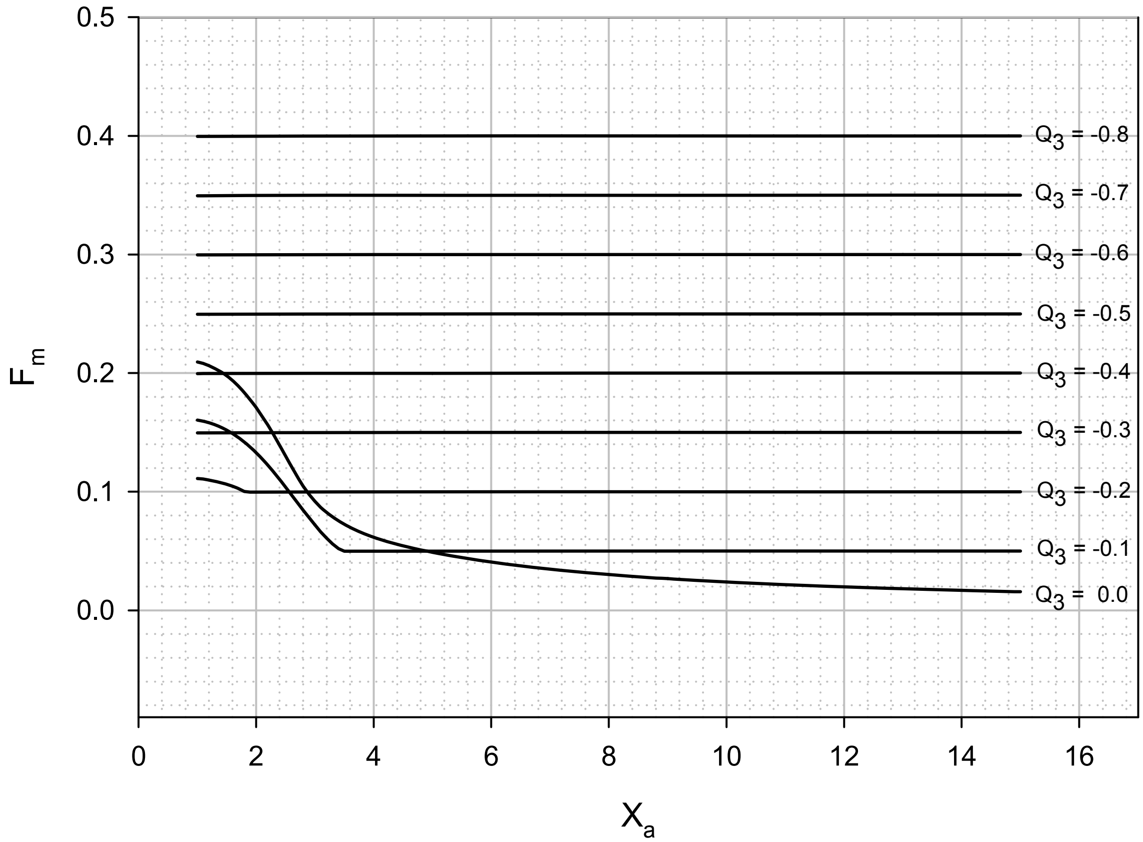
Figure 4.18.6 – Z_d, Z_v, Z_w, and Z_m Verus X_a



Notes:

1. The curves for F_m are valid for $\nu^* = 0.4$. These curves sufficiently accurate for other values of ν^*.
2. If X_a and Q_3 are beyond the values in the curves, F_m shall be calculated using Table 4.18.3.

Figure 4.18.7 – F_m Verus X_a $\left(0.0 \le Q_3 \le 0.8\right)$



Notes:

1. The curves for F_m are valid for $\nu^* = 0.4$. These curves sufficiently accurate for other values of ν^*.
2. If X_a and Q_3 are beyond the values in the curves, F_m shall be calculated using Table 4.18.3.

Figure 4.18.8 – F_m Verus X_a $\left(-0.8 \le Q_3 \le 0.0\right)$

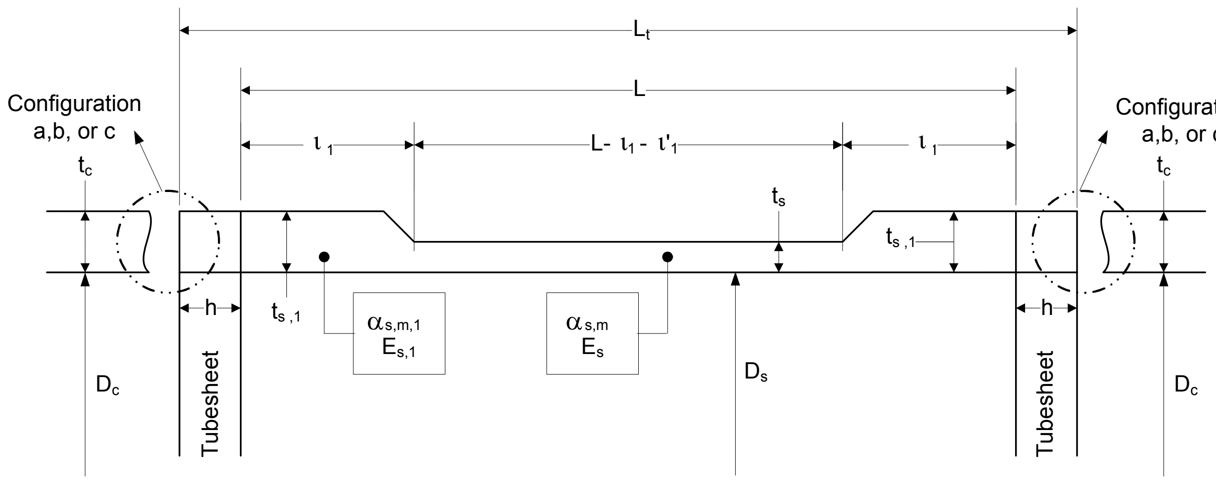
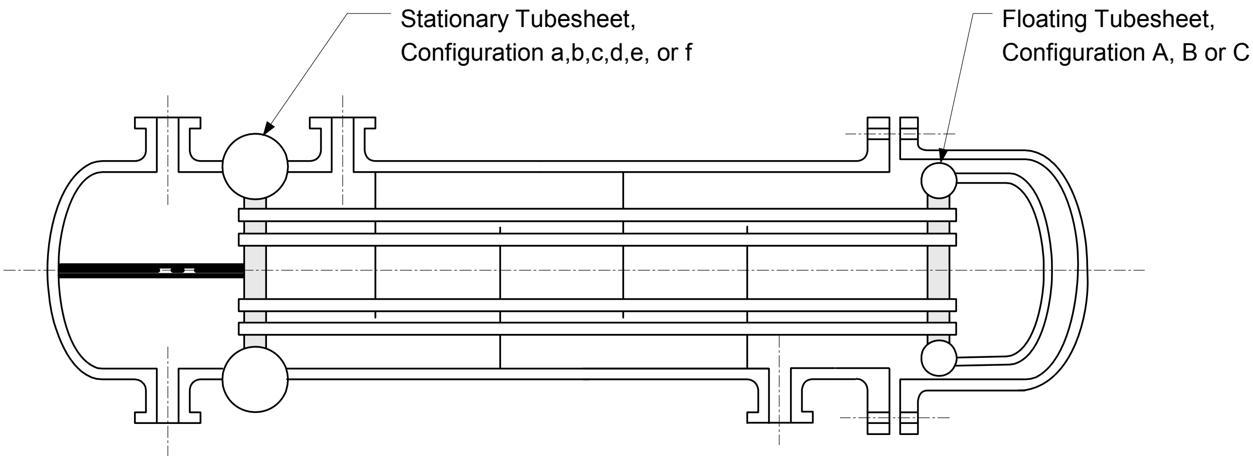
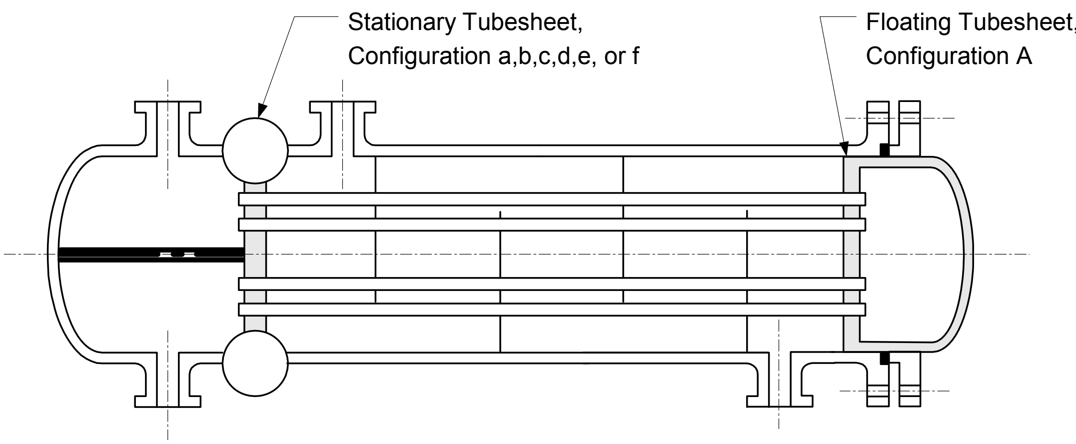


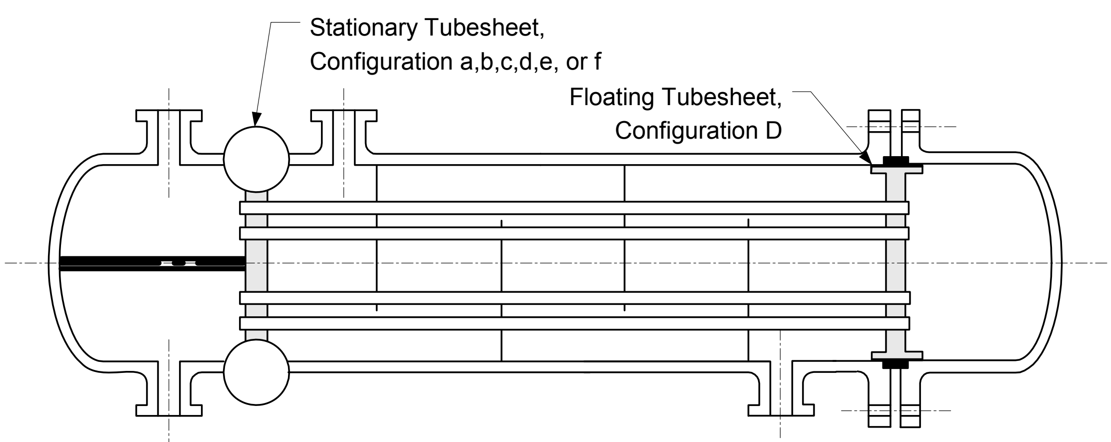
Figure 4.18.9 – Shell With Increased Thickness Adjacent to the Tubesheets



(a) Typical Floating Tubesheet Exchanger With An Immersed Floating Head

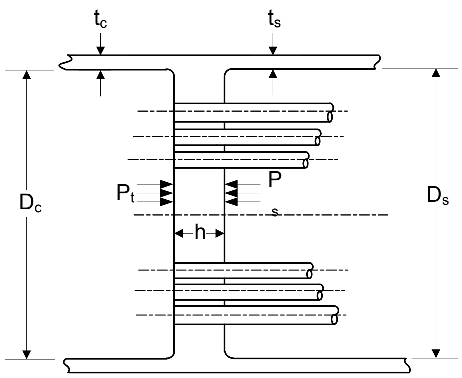


(b) Typical Floating Tubesheet Exchanger With An Externally Sealed Floating Head

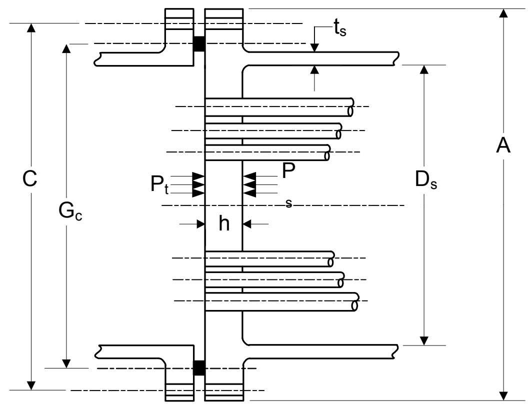


(c) Typical Floating Tubesheet Exchanger With An Internally Sealed Floating Tubesheet

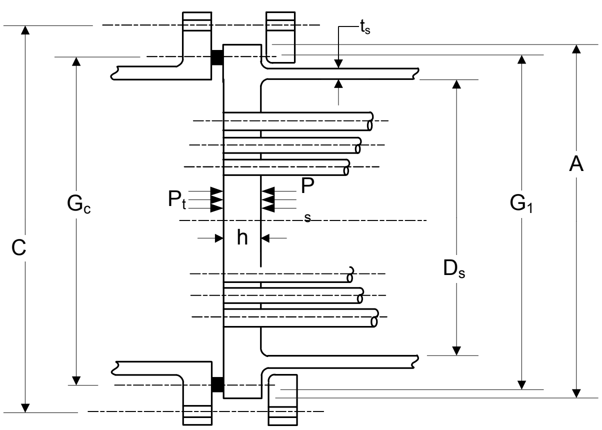
Figure 4.18.10 – Floating Tubesheet Heat Exchangers



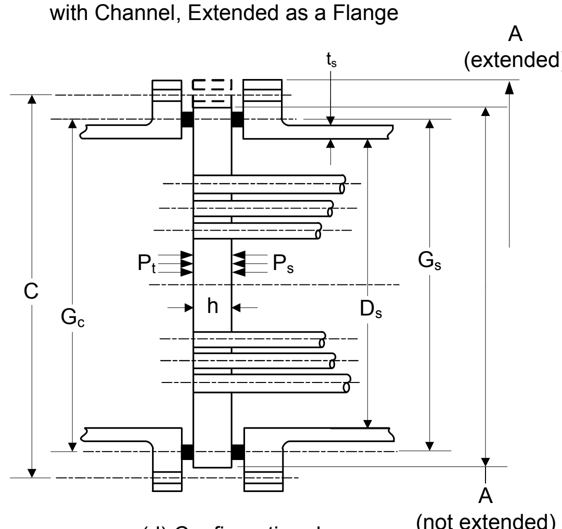
(a) Configuration a:
Tubesheet Integral with Shell and Channel



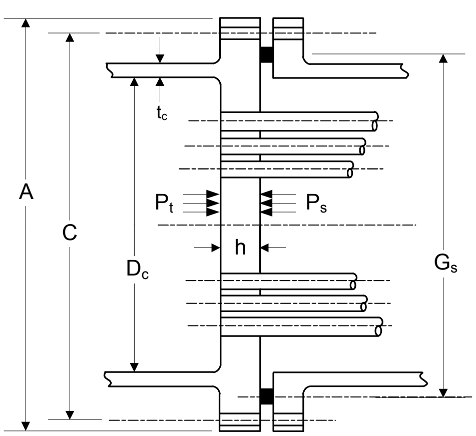
(b) Configuration b:
Tubesheet Integral with Shell and Gasketed
with Channel, Extended as a Flange



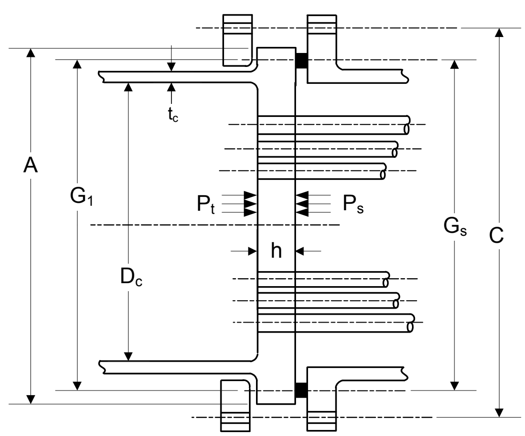
(c) Configuration c:
Tubesheet Integral with Shell and Gasketed
with Channel, Not Extended as a Flange



(d) Configuration d:
Tubesheet Integral with Shell and Channel

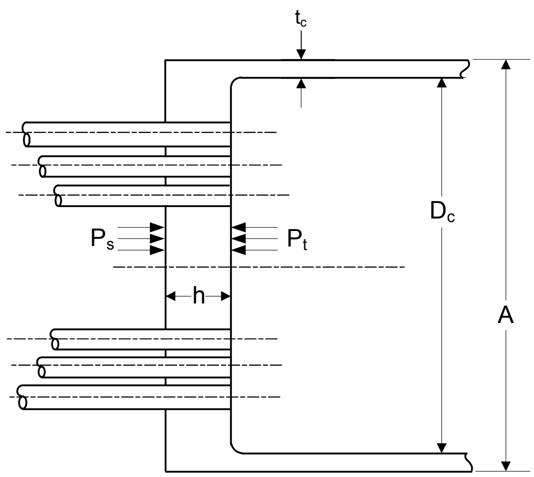


(e) Configuration e:
Tubesheet Gasketed with Shell and Integral with
Channel, Extended as a Flange

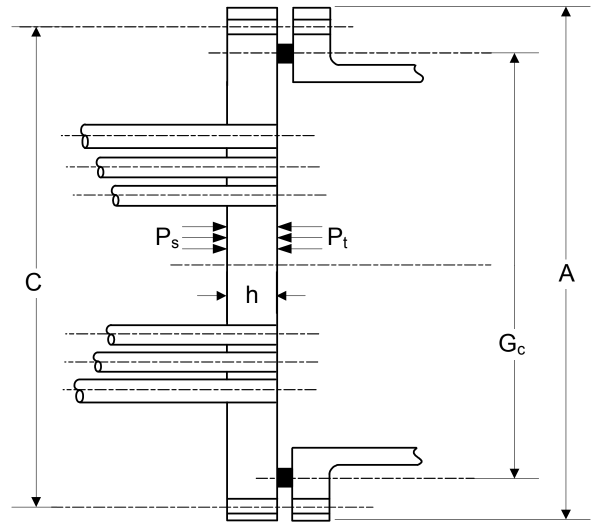


(f) Configuration f:
Tubesheet Gasketed with Shell and Integral with
Channel, Not Extended as a Flange

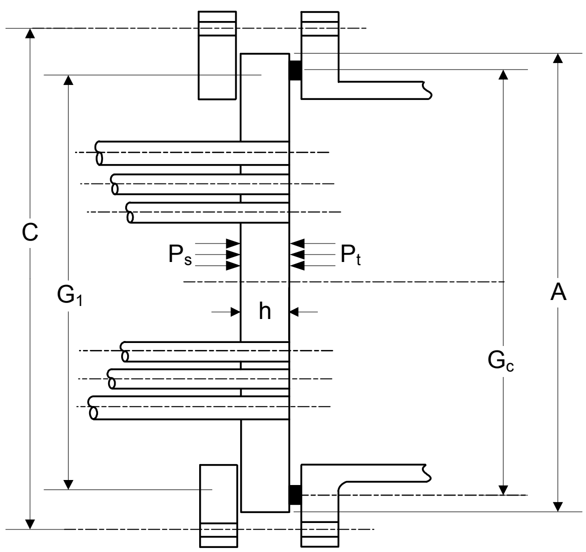
Figure 4.18.11 – Stationary Tubesheet Configurations



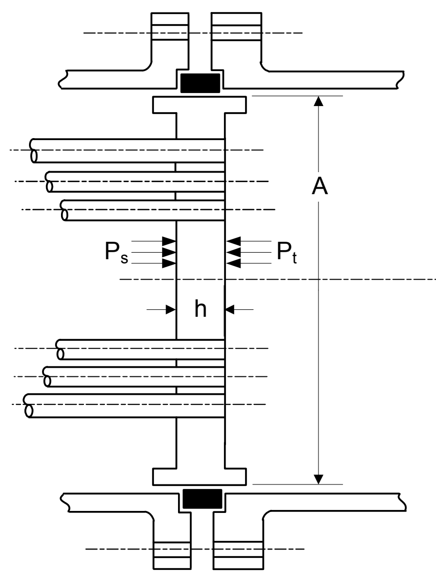
(a) Configuration A:
Tubesheet Integral



(b) Configuration B:
Tubesheet Gasketed, Extended As A Flange



(c) Configuration C:
Tubesheet Gasketed, Not Extended As A Flange



(d) Configuration D:
Tubesheet Internally Sealed

Figure 4.18.12 – Floating Tubesheet Configurations

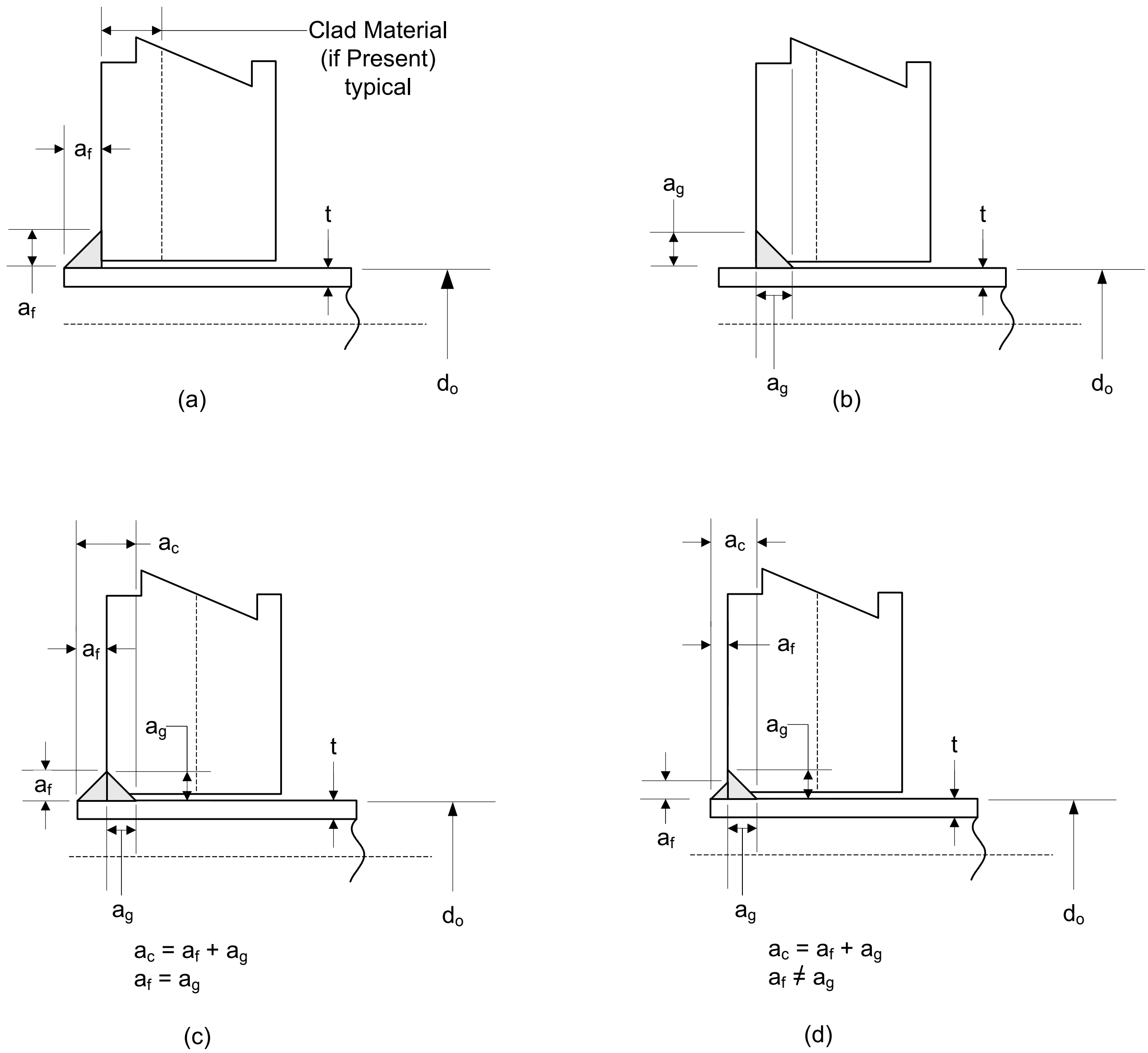
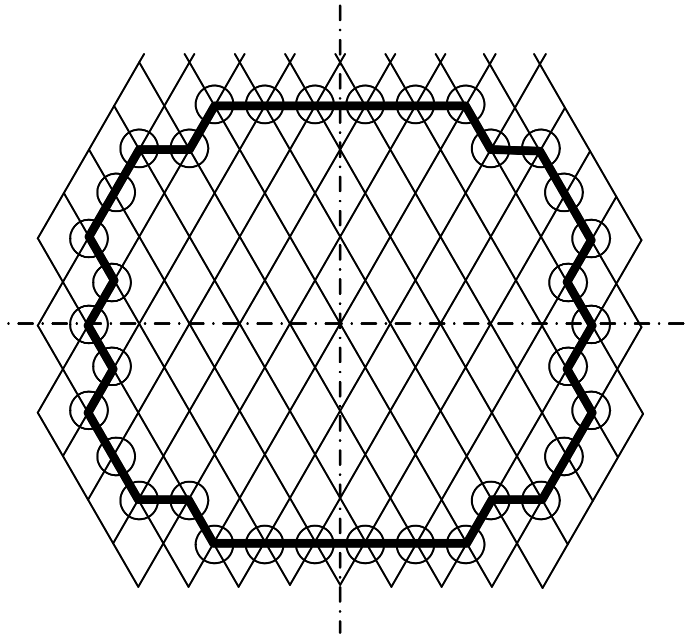
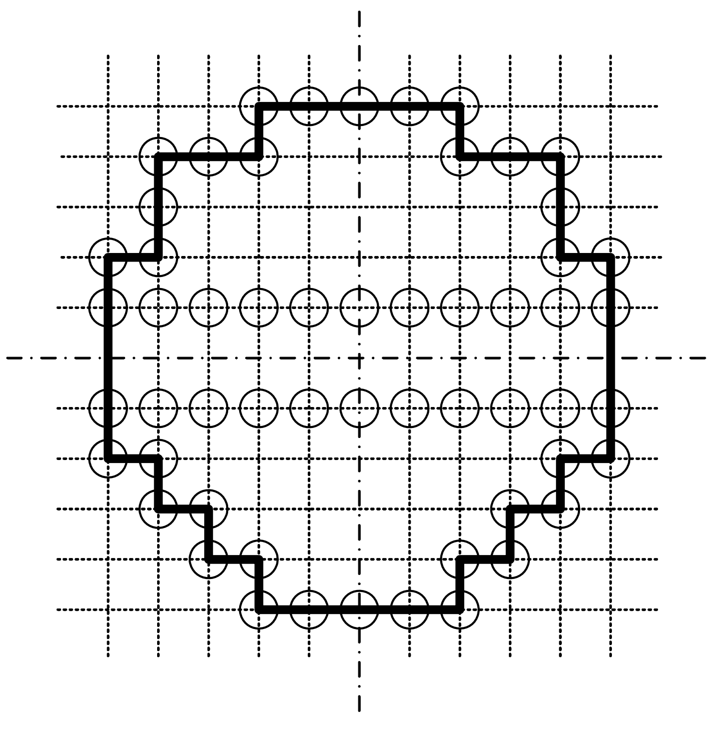


Figure 4.18.13 – Some Acceptable Types of Tube-To-Tubesheet Strength Welds



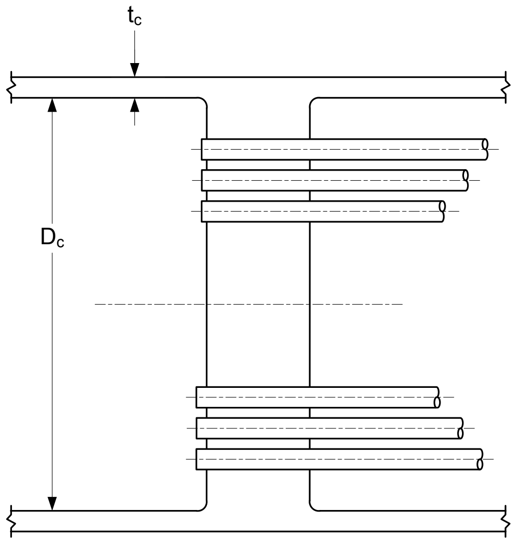
(a) Equilateral Triangular Pattern



(b) Square Pattern

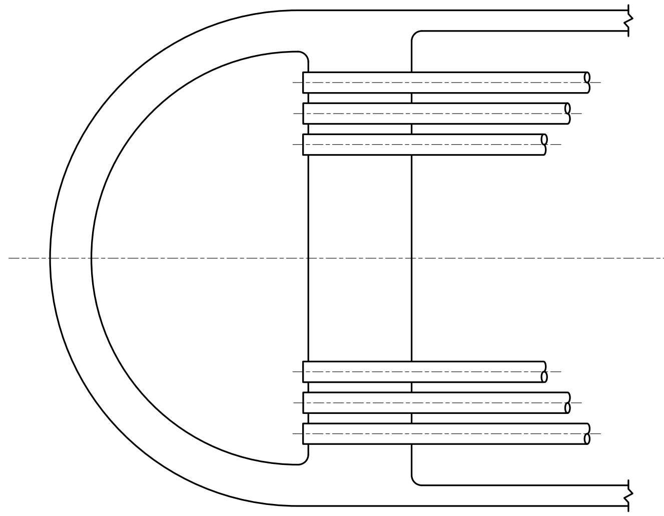
Note: C_p (Perimeter) is the length of the heavy line

Figure 4.18.14 – Tube Layout Perimeter



(a) Cylindrical Channel

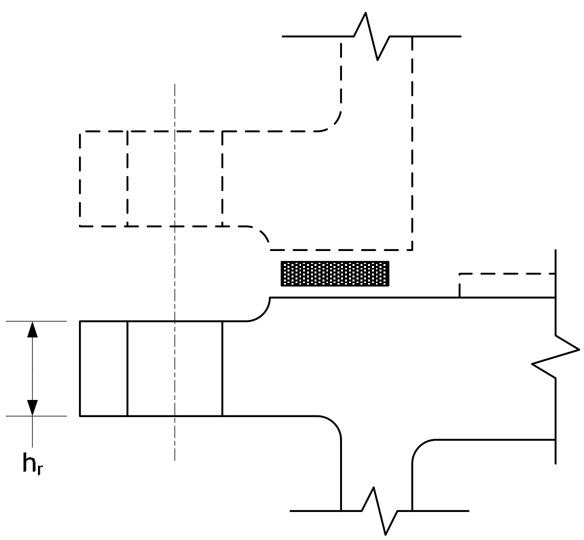
NOTE: Length of Cylinder Shall Be $\geq 1.8\sqrt{D_c t_c}$



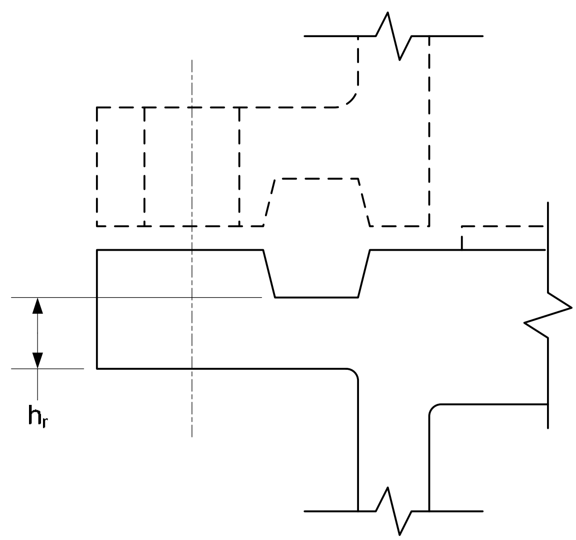
(b) Hemispherical Channel

NOTE: Head Shall Be 180 Degrees With No Intervening Cylinders

Figure 4.18.15 - Integral Channels



(a) Raised Face



(b) Grooved for Ring Gasket

Figure 4.18.16 - Some Representative Configurations Describing the Minimum Required Thickness of the Tubesheet Flanged Extension, h_r

4.19 Design Rules for Bellows Expansion Joints

4.19.1 Scope

The rules in paragraph 4.19 apply to single or multiple layer bellows expansion joints, unreinforced, reinforced or toroidal, as shown in Figure 4.19.1, subject to internal or external pressure and cyclic displacement. The bellows shall consist of single or multiple identically formed convolutions. They may be as formed (not heat-treated), or annealed (heat-treated). The suitability of an expansion joint for the specified design pressure, temperature, and axial displacement shall be determined by the methods described herein.

4.19.2 Conditions of Applicability

The design rules of this paragraph are applicable only when the following conditions of applicability are satisfied:

a) The bellows length shall be such that: $Nq \leq 3D_b$.

b) The bellows nominal thickness shall be such that: $nt \leq 5mm\ (0.2\,in.)$.

c) The number of plies shall be such that: $n \leq 5$.

d) The displacement shall be essentially axial. However angular and/or lateral deflection inherent in the fit-up of the expansion joint to the pressure vessel is permissible provided the amount is specified and is included in the expansion joint design (see paragraph 4.19.3.1.d).

e) These rules are valid for design temperatures up to 425°C (800°F).

f) The fatigue equations given in Table 4.19.5, Table 4.19.7 and Table 4.19.10 are valid for austenitic chromium-nickel steels, UNS N066XX and UNS N04400. For other materials, the fatigue evaluation shall meet the requirements of paragraph 4.19.3.2.b.

4.19.3 Design Considerations

4.19.3.1 General

a) Expansion joints used as an integral part of heat exchangers or other pressure vessels shall be designed to provide flexibility for thermal expansion and also to function as a pressure containing element.

b) The vessel Manufacturer shall specify the design conditions and requirements for the detailed design and manufacture of the expansion joint. Use of the specification sheet shown in paragraph 4.19.17 is recommended.

c) In all vessels with integral expansion joints, the hydrostatic end force caused by pressure and/or the joint spring force shall be resisted by adequate restraint elements (e.g., exchanger tubes or shell, external restraints, anchors, etc.). The primary stress in these restraining elements shall satisfy paragraph 4.1.6.1.

d) The expansion joints shall be provided with bars or other suitable members for maintaining the proper overall length dimension during shipment and vessel fabrication. Expansion bellows shall not be extended, compressed, rotated, or laterally offset to accommodate connecting parts, which are not properly aligned, unless the design considers such movements (see paragraph 4.19.8).

e) The minimum thickness limitations of paragraph 4.1 do not apply to bellows designed to this paragraph.

f) This paragraph does not contain rules to cover all details of design and construction. The criteria in this paragraph are, therefore, established to cover common expansion joint types, but it is not intended to limit configurations or details to those illustrated or otherwise described herein. However, when evaluating designs which differ from the basic concepts of this paragraph (e.g., asymmetric geometries or loadings, external pressure, materials, etc.), the design shall comply with the requirements of Part 5.

g) Longitudinal weld seams that comply with paragraphs 4.19.9 and 4.19.10 shall be considered to have a joint efficiency of 1.0.

h) The elastic moduli, yield strength, and allowable stresses shall be taken at the design temperatures. However, when performing the fatigue evaluation in accordance with 4.19.5.7 (unreinforced bellows), 4.19.6.7 (reinforced bellows) and 4.19.7.7 (toroidal bellows), it is permitted to use the operating metal temperature instead of design temperature.

4.19.3.2 Fatigue

a) Cumulative Damage – If there are two or more types of stress cycles, which produce significant stresses, their cumulative effect shall be evaluated as given below.

1) Designate the specified number of times each type of stress cycle of types 1, 2, 3, etc., will be repeated during the life of the expansion joint as $n_1,\ n_2,\ n_3$, etc., respectively. When determining $n_1,\ n_2,\ n_3$, etc., consideration shall be given to the superposition of cycles of various origins, which produce a total stress difference range greater than the stress difference ranges of the individual cycles. For example, if one type of stress cycle produces 1000 cycles of a stress difference variation from zero to +60,000 psi and another type of stress cycle produces 10,000 cycles of a stress difference variation from zero to -50,000 psi, the two types of cycles to be considered are defined by the following parameters:

i) Type 1 cycle: $n_1 = 1000;\ S_{t1} = |60000| + |-50000| = 110,000\ psi$

ii) Type 2 cycle: $n_2 = 10000 - 1000 = 9000;\ S_{t2} = |0| + |-50000| = 50,000\ psi$

2) For each value $S_{t1},\ S_{t2},\ S_{t3}$, etc., use the applicable design fatigue curve to determine the maximum number of repetitions which would be allowable if this type of cycle were the only one acting. Call these values $N_1,\ N_2,\ N_3$, etc.

3) For each type of stress cycle, calculate the usage factors U_1, U_2, U_3 etc., from $U_1 = n_1/N_1$, $U_2 = n_2/N_2$, $U_3 = n_3/N_3$, or for the ith cycle:

$$U_i = \frac{n_i}{N_i} \tag{4.19.1}$$

4) Calculate the cumulative usage factor U by summing the individual factors, or $U = U_1 + U_2 + U_3 + ... + U_m$.

5) The cumulative usage factor U for the total number of stress cycles, m, shall not exceed 1.0, or

$$\sum_{i=1}^{m} U_i = \sum_{i=1}^{m} \frac{n_i}{N_i} \le 1.0 \tag{4.19.2}$$

b) Fatigue Correlation Testing – In complying with the requirements of paragraph 4.19.5.7 (unreinforced bellows), paragraph 4.19.6.7 (reinforced bellows), or paragraph 4.19.7.7 (toroidal bellows) the calculation and relation to fatigue life may be performed by any method based on the theory of elasticity. However, the method must be substantiated by correlation with proof or strain gage testing (see Annex 5.E) on a consistent series of bellows of the same basic design (annealed, and as formed bellows are considered as separate designs) by the bellows Manufacturer to demonstrate predictability of rupture pressure and cyclic life.

1) The substantiation of any analytical procedure shall be based on data obtained from five separate tests on bellows of the same basic design. When substantiating bellows designs with more than two convolutions in series, the test data shall have been obtained from bellows with a minimum of three convolutions.

2) When compared with the data obtained from the calculation procedure, the test data shall demonstrate that the rupture pressure of the bellows is equal to or greater than three times the maximum allowable working pressure at room temperature.

3) When S_t along with the other appropriate factors are used in the cycle life equations in paragraph 4.19.5.7 (unreinforced bellows), paragraph 4.19.6.7 (reinforced bellows), or paragraph 4.19.7.7 (toroidal bellows), the required design life N_{alw} shall be less than the calculated cycles to failure based on the data obtained by testing. The design cycle life may not be increased above that obtained from the equations in these paragraphs regardless of the test results. The substantiation of analytical procedures shall be available for review by the Inspector.

4.19.4 Materials

Pressure-restraining component materials including the restraining elements covered by paragraph 4.19.3.1.c shall comply with the requirements of Part 3.

4.19.5 Design of U-Shaped Unreinforced Bellows

4.19.5.1 Scope

These rules cover the design of bellows having unreinforced U-shaped convolutions. Each half convolution consists of a sidewall and two tori of nearly the same radius (at the crest and root of the convolution), in the neutral position, so that the convolution profile presents a smooth geometrical shape as shown in Figure 4.19.1 Sketch (a).

4.19.5.2 Conditions of Applicability

The following conditions of applicability apply in addition to those listed in paragraph 4.19.2.

a) A variation of 10% between the crest convolution radius r_{ic} and the root convolution radius r_{ir} is permitted (see Figure 4.19.2)

b) The torus radius shall be such that: $r_i \geq 3t$, where: $r_i = (r_{ic} + r_{ir})/2$,

c) The off-set angle of the sidewalls, α , in the neutral position shall be such that: $-15^o \leq \alpha \leq +15^o$ degrees (see Figure 4.19.2). In this case, q is defined as the length between two consecutive convolutions when their sidewalls have been made parallel.

d) The convolution height shall be such that: $w \leq D_b/3$.

4.19.5.3 Internal Pressure

The required stress calculations and acceptance criteria for U-shaped unreinforced bellows are given in Table 4.19.1.

4.19.5.4 Column Instability Due to Internal Pressure

a) The allowable internal design pressure to avoid column instability is given by the following equation.

$$P_{sc} = \frac{0.34\pi K_b}{Nq} \tag{4.19.3}$$

b) The internal pressure shall satisfy the following equation.

$$P \le P_{sc} \tag{4.19.4}$$

4.19.5.5 In-Plane Instability Due to Internal Pressure

a) The allowable internal design pressure based on in-plane instability is given by the following equation.

$$P_{si} = \frac{AS_y^*(\pi - 2)}{D_m q \left[1 + 2\delta^2 + \left(1 - 2\delta^2 + 4\delta^4\right)^{0.5}\right]^{0.5}} \tag{4.19.5}$$

where

$$\delta = \frac{S_4}{3S_{2,I}} \tag{4.19.6}$$

b) S_y^* is the effective yield strength at design temperature (unless otherwise specified) of bellows material in the as-formed or annealed condition. If S_y^* is not available in material standards, the values shown in the following equations shall be used where S_y is the yield strength of bellows material at design temperature, given by Annex 3.D. Higher values of S_y^* may be used if justified by representative tests.

$$S_y^* = 2.3S_y \qquad \textit{for as-formed bellows} \tag{4.19.7}$$

$$S_y^* = 0.75S_y \qquad \textit{for annealed bellows} \tag{4.19.8}$$

c) The internal pressure shall satisfy the following equation.

$$P \le P_{si} \tag{4.19.9}$$

4.19.5.6 External Pressure Strength

a) External pressure capacity – The rules of paragraph 4.19.5.3 shall be applied taking P as the absolute value of the external pressure. When the expansion bellows is subjected to vacuum, the design shall be performed assuming that only the internal ply resists the pressure. The pressure stress equations of paragraph 4.19.5.3 shall be applied with $n = 1$.

b) Instability due to external pressure – The design shall be performed according to the rules of paragraph 4.4 by replacing the bellows with an equivalent cylinder, using an equivalent outside diameter, D_{eq}, and an equivalent thickness, e_{eq}, given by the following equations where I_{xx} is the moment of inertia (see Figure 4.19.3).

$$D_{eq} = D_b + w + 2e_{eq} \tag{4.19.10}$$

$$e_{eq} = \left(\frac{12\left(1 - \nu_b^2\right) I_{xx}}{q}\right)^{1/3} \tag{4.19.11}$$

c) If $L_t = 0$, I_{xx} is given by the following equation.

$$I_{xx} = nt_p \left[\frac{(2w-q)^3}{48} + 0.4q(w-0.2q)^2 \right] \tag{4.19.12}$$

4.19.5.7 Fatigue Evaluation

a) The meridional membrane and bending stresses due to the total equivalent axial displacement range Δq of each convolution is given by the following equations.

$$S_5 = \frac{E_b t_p^2 \Delta q}{2w^3 C_f} \tag{4.19.13}$$

$$S_6 = \frac{5E_b\, t_p \Delta q}{3w^2 C_d} \tag{4.19.14}$$

C_f and C_d are evaluated using Tables 4.19.3 and 4.19.4, respectively.

b) The total stress range due to cyclic displacement is given by the following equation.

$$S_t = 0.7\left[S_3 + S_4\right] + \left[S_5 + S_6\right] \tag{4.19.15}$$

c) Calculation of allowable number of cycles

1) The specified number of cycles N_{spe} shall be specified in consideration of the anticipated number of cycles expected to occur during the operating life of the bellows. The allowable number of cycles shall satisfy the following equation.

$$N_{alw} \geq N_{spe} \tag{4.19.16}$$

2) The allowable number of cycles, N_{alw}, shall be calculated using the equations in Table 4.19.5. These equations are valid for:

   i) Austenitic chromium-nickel stainless steels, UNS N066XX and UNS N04400 for metal temperatures not exceeding 425°C (800°F).

   ii) U-shaped unreinforced bellows, as-formed or annealed

3) If the bellows is subjected to different cycles of pressure or displacement, such as those produced by start-up or shutdown, their cumulative damage shall be considered as in paragraph 4.19.3.2.a.

4) For fatigue correlation testing, see paragraph 4.19.3.2.b.

4.19.5.8 Axial Stiffness

The theoretical axial stiffness of a bellows comprised of N convolutions may be evaluated by the following equation. This equation is valid only in the elastic range. Outside of the elastic range lower values can be used, based upon Manufacturer's experience or representative test results.

$$K_b = \frac{\pi E_b\, D_m}{2\left(1-\nu_b^2\right)C_f}\left(\frac{n}{N}\right)\left(\frac{t_p}{w}\right)^3 \tag{4.19.17}$$

4.19.6 Design of U-Shaped Reinforced Bellows

4.19.6.1 Scope

These rules cover the design of bellows having U-shaped convolutions with rings to reinforce the bellows against internal pressure. Each half convolution consists of a sidewall and two tori of nearly the same radius (at the crest and root of the convolution), in the neutral position, so that the convolution profile presents a smooth geometrical shape as shown in Figure 4.19.1 Sketch (b).

4.19.6.2 Conditions of Applicability

The conditions of applicability are given in paragraph 4.19.5.2.

4.19.6.3 Internal Pressure

The required stress calculations and acceptance criteria for U-shaped reinforced bellows are given in Table 4.19.6.

4.19.6.4 Column Instability Due to Internal Pressure

a) The allowable internal design pressure to avoid column instability is given by the following equation.

$$P_{sc} = \frac{0.3\pi K_b}{Nq} \tag{4.19.18}$$

b) The internal pressure shall satisfy the following equation.

$$P \le P_{sc} \tag{4.19.19}$$

4.19.6.5 In-Plane Instability Due to Internal Pressure

Reinforced bellows are not prone to in-plane instability.

4.19.6.6 External Pressure Strength

a) External pressure capacity – The rules of paragraph 4.19.5.3 relative to unreinforced bellows shall be applied taking P as the absolute value of the external pressure. When the expansion bellows is subjected to vacuum, the design shall be performed assuming that only the internal ply resists the pressure. The pressure stress equations of paragraph 4.19.5.3 shall be applied with $n = 1$.

b) Instability due to external pressure – The circumferential instability of a reinforced bellows shall be calculated in the same manner as for unreinforced bellows (see paragraph 4.19.5.6.b)

4.19.6.7 Fatigue Evaluation

a) The meridional membrane and bending stresses due to the total equivalent axial displacement range Δq of each convolution are given by the following equations.

$$S_5 = \frac{E_b t_p^2 \Delta q}{2\left(w - C_r q\right)^3 C_f} \tag{4.19.20}$$

$$S_6 = \frac{5E_b t_p \Delta q}{3\left(w - C_r q\right)^2 C_d} \tag{4.19.21}$$

C_f and C_d are evaluated using Tables 4.19.3 and 4.19.4, respectively.

b) The total stress range due to cyclic displacement is given by Equation (4.19.15).

c) The fatigue evaluation of a reinforced bellows shall be calculated in the same manner as for unreinforced bellows (see paragraph 4.19.5.7.c), except that the allowable number of cycles shall be calculated using the equations in Table 4.19.7.

4.19.6.8 Axial Stiffness

The theoretical axial stiffness of a bellows comprised of N convolutions is given by the following equation.

$$K_b = \frac{\pi E_b D_m}{2\left(1-\nu_b^2\right)C_f}\left(\frac{n}{N}\right)\left(\frac{t_p}{w-C_r q}\right)^3 \tag{4.19.22}$$

This equation is valid only in the elastic range. Outside of the elastic range lower values can be used, based upon Manufacturer's experience or representative test results.

4.19.7 Design of Toroidal Bellows

4.19.7.1 Scope

These rules cover the design of bellows having toroidal convolutions. Each convolution consists of a torus of radius r as shown in Figure 4.19.1 Sketch (c).

4.19.7.2 Conditions of Applicability

The conditions of applicability are given in paragraph 4.19.2.

4.19.7.3 Internal Pressure

The required stress calculations and acceptance criteria for toroidal bellows are given in Table 4.19.8.

4.19.7.4 Column Instability Due to Internal Pressure

a) The allowable internal design pressure to avoid column instability is given by the following equation.

$$P_{sc} = \frac{0.15\pi K_b}{Nr} \tag{4.19.23}$$

b) The internal pressure shall satisfy the following equation.

$$P \le P_{sc} \tag{4.19.24}$$

4.19.7.5 In-Plane Instability Due to Internal Pressure

Toroidal bellows are not prone to in-plane instability.

4.19.7.6 External Pressure Strength

a) External pressure capacity – The rules of paragraph 4.19.7.3 shall be applied taking P as the absolute value of the external pressure and using $A_c = 0$ in the equations. When the expansion bellows is subjected to vacuum, the design shall be performed assuming that only the internal ply resists the pressure. The pressure stress equations of paragraph 4.19.7.3 shall be applied with $n = 1$.

b) Instability due to external pressure is not covered by the present rules.

4.19.7.7 Fatigue Evaluation

a) The meridional membrane and bending stresses due to the total equivalent axial displacement range Δq of each convolution are given by the following equations.

$$S_5 = \frac{E_b t_p^2 B_1 \Delta q}{34.3 r^3} \tag{4.19.25}$$

$$S_6 = \frac{E_b t_p B_2 \Delta q}{5.72 r^2} \tag{4.19.26}$$

B_1 and B_2 are evaluated using Table 4.19.9.

b) The total stress range due to cyclic displacement is given by the following equation.

$$S_t = 3S_3 + S_5 + S_6 \tag{4.19.27}$$

c) The fatigue evaluation of a toroidal bellows shall be calculated in the same manner as for unreinforced bellows (see paragraph 4.19.5.7.c), except that the allowable number of cycles shall be calculated using the equations in Table 4.19.10.

4.19.7.8 Axial Stiffness

The theoretical axial stiffness of a bellows comprised of N convolutions is given by the following equation.

$$K_b = \frac{E_b D_m B_3}{12\,(1-\nu_b^{\,2})}\left(\frac{n}{N}\right)\left(\frac{t_p}{r}\right)^3 \tag{4.19.28}$$

B_3 is evaluated using Table 4.19.9. This equation is valid only in the elastic range. Outside of the elastic range lower values can be used, based upon Manufacturer's experience or representative test results.

4.19.8 Bellows Subject to Axial, Lateral or Angular Displacements

4.19.8.1 General

The purpose of this paragraph is to determine the equivalent axial displacement of an expansion bellows subjected at its ends to:

a) an axial displacement from the neutral position: x in extension $(x > 0)$ or in compression $(x < 0)$

b) a lateral deflection from the neutral position: y, $(y > 0)$

c) an angular rotation from the neutral position: θ, $(\theta > 0)$

4.19.8.2 Axial displacement

a) When the ends of the bellows are subjected to an axial displacement x (see Figure 4.19.4), the equivalent axial displacement per convolution is given by the following equation. In this equation, x shall be taken as positive for extension $(x > 0)$ and negative for compression $(x < 0)$. Values of x in extension and compression may be different.

$$\Delta q_x = \frac{x}{N} \tag{4.19.29}$$

b) The corresponding axial force F_x applied to the ends of the bellows is given by the following equation.

$$F_x = K_b x \tag{4.19.30}$$

4.19.8.3 Lateral deflection

a) When the ends of the bellows are subjected to a lateral deflection y (see Figure 4.19.5), the equivalent axial displacement per convolution is given by the following equation where y shall be taken positive.

$$\Delta q_y = \frac{3\, D_m y}{N\left(Nq + x\right)} \tag{4.19.31}$$

b) The corresponding lateral force F_y applied to the ends of the bellows is given by the following equation.

$$F_y = \frac{3 K_b D_m^2 y}{2\left(Nq + x\right)^2} \tag{4.19.32}$$

c) The corresponding moment M_y applied to the ends of the bellows is given by the following equation.

$$M_y = \frac{3 K_b D_m^2 y}{4\left(Nq + x\right)} \tag{4.19.33}$$

4.19.8.4 Angular rotation

a) When the ends of the bellows are subjected to an angular rotation θ (see Figure 4.19.6), the equivalent axial displacement per convolution is given by the following equation where θ, expressed in radians, shall be taken positive.

$$\Delta q_\theta = \frac{D_m \theta}{2\, N} \tag{4.19.34}$$

b) The corresponding moment M_θ applied to the ends of the bellows is given by the following equation.

$$M_\theta = \frac{K_b D_{\mathrm{m}}^2 \theta}{8} \tag{4.19.35}$$

4.19.8.5 Total equivalent axial displacement range per convolution

a) Equivalent axial displacement per convolution – The equivalent axial displacement per convolution, in extension or compression, is given by the following equations.

$$\Delta q_{\mathrm{e}} = \Delta q_{\mathrm{x}} + \Delta q_{\mathrm{y}} + \Delta q_\theta \qquad (\mathit{extended\ convolution}) \tag{4.19.36}$$

$$\Delta q_{\mathrm{c}} = \Delta q_{\mathrm{x}} - \Delta q_{\mathrm{y}} - \Delta q_\theta \qquad (\mathit{compressed\ convolution}) \tag{4.19.37}$$

b) Bellows installed without cold spring – If the bellows is subjected to displacements from the neutral position $\left(x_0 = 0\ ,\ y_0 = 0\ ,\ \theta_0 = 0\right)$ to the operating position $\left(x, y, \theta\right)$ (see Figure 4.19.7), the

equivalent axial displacement per convolution, in extension or compression, for the initial and operating positions and the total equivalent axial displacement range are given by the following equations.

Initial Position:

$$\Delta q_{\mathrm{e}} = 0.0 \qquad (extension) \tag{4.19.38}$$

$$\Delta q_{\mathrm{c}} = 0.0 \qquad (compression) \tag{4.19.39}$$

Operating Position:

$$\Delta q_{\mathrm{e}} = \Delta q_{\mathrm{x}} + \Delta q_{\mathrm{y}} + \Delta q_{\theta} \qquad (extension) \tag{4.19.40}$$

$$\Delta q_{\mathrm{c}} = \Delta q_{\mathrm{x}} - \Delta q_{\mathrm{y}} - \Delta q_{\theta} \qquad (compression) \tag{4.19.41}$$

Total Equivalent Axial Displacement Range:

$$\Delta q = \max\left[\left|\Delta q_e\right|,\ \left|\Delta q_c\right|\right] \tag{4.19.42}$$

c) Bellows installed with cold spring – If the bellows is subjected to displacements from an initial position $\left(x_0,\ y_0,\ \theta_0\right)$, which is not the neutral position to the operating position $\left(x, y, \theta\right)$ (see Figure 4.19.8), the equivalent axial displacement per convolution, in extension or compression, for the initial and operating positions and the total equivalent axial displacement range are given by the following equations.

Initial Position:

$$\Delta q_{\mathrm{e},0} = \Delta q_{\mathrm{x},0} + \Delta q_{\mathrm{y},0} + \Delta q_{\theta,0} \qquad (extension) \tag{4.19.43}$$

$$\Delta q_{c,0} = \Delta q_{x,0} - \Delta q_{y,0} - \Delta q_{\theta,0} \qquad (compression) \tag{4.19.44}$$

Operating Position:

$$\Delta q_{\mathrm{e}} = \Delta q_{\mathrm{x}} + \Delta q_{\mathrm{y}} + \Delta q_{\theta} \qquad (extension) \tag{4.19.45}$$

$$\Delta q_{c} = \Delta q_{x} - \Delta q_{y} - \Delta q_{\theta} \qquad (compression) \tag{4.19.46}$$

Total Equivalent Axial Displacement Range:

$$\Delta q = \max\left[\left|\Delta q_{\mathrm{e}} - \Delta q_{\mathrm{c},0}\right|,\ \left|\Delta q_{\mathrm{c}} - \Delta q_{\mathrm{e},0}\right|\right] \tag{4.19.47}$$

d) Bellows operating between two operating positions – If the bellows is subjected to displacements from operating position number 1 $\left(x_1,\ y_1,\ \theta_1\right)$ to the operating position number 2 $\left(x_2,\ y_2,\ \theta_2\right)$ (see Figure 4.19.9), the equivalent axial displacement per convolution, in extension or compression, for operating positions number 1 and 2 and the total equivalent axial displacement range are given by the following equations. An initial cold spring (initial position 0) has no effect on the results.

Position Number 1:

$$\Delta q_{e,1} = \Delta q_{x,1} + \Delta q_{y,1} + \Delta q_{\theta,1} \qquad (extension) \tag{4.19.48}$$

$$\Delta q_{c,1} = \Delta q_{x,1} - \Delta q_{y,1} - \Delta q_{\theta,1} \qquad (compression) \tag{4.19.49}$$

Position Number 2:

$$\Delta q_{e,2} = \Delta q_{x,2} + \Delta q_{y,2} + \Delta q_{\theta,2} \qquad (extension) \tag{4.19.50}$$

$$\Delta q_{c,2} = \Delta q_{x,2} - \Delta q_{y,2} - \Delta q_{\theta,2} \qquad (compression) \tag{4.19.51}$$

Total Equivalent Axial Displacement Range:

$$\Delta q = \max\left[\ \left|\Delta q_{e,2} - \Delta q_{c,1}\right|, \ \left|\Delta q_{c,2} - \Delta q_{e,1}\right| \ \right] \tag{4.19.52}$$

4.19.9 Fabrication

The following requirements shall be met in the fabrication of expansion joint flexible elements.

a) All welded joints shall comply with the requirements of Part 6.

b) All longitudinal weld seams shall be butt-type full penetration welds; Type 1 welds of paragraph 4.2.

c) Bellows shall be attached to the weld end elements by circumferential butt or full fillet welds as shown in Figure 4.19.10.

d) Other than the attachment welds, no circumferential welds are permitted in the fabrication of bellows convolutions.

4.19.10 Examination

The following examinations are required to verify the integrity of expansion joints.

a) All expansion joint flexible elements shall be visually examined for and shall be free of injurious defects, such as notches, crevices, material buildup or upsetting, weld spatter, etc. which may serve as points of local stress concentrations. Suspect surface areas shall be further examined by liquid penetrant.

b) All bellows butt-type welds shall be examined 100 % on the inside and outside surfaces by the liquid penetrant method before forming. This examination shall be repeated after forming to the maximum extent possible considering the physical and visual access to the weld surfaces after forming. The butt weld shall be full penetration.

c) The circumferential attachment welds between the bellows and the weld ends shall be examined 100 % by liquid penetrant.

d) Liquid penetrant examination shall be in accordance with Part 7. However, any linear indication found by examination shall be considered relevant if the dimension exceeds $0.25t_m$, but not less than 0.25 mm (0.010 in.), where t_m is the minimum bellows wall thickness before forming.

4.19.11 Pressure Test Requirements

The pressure testing requirements for expansion joints shall be as follows.

a) The completed expansion joint shall be subjected to a pressure test in accordance with paragraph 4.1.6.2 and Part 8. However the designer shall consider the possibility of instability of the bellows due to internal pressure if the test pressure exceeds the value determined using the following equation. In such a case, the designer shall redesign the bellows to satisfy the test condition.

$$P_{t,s} = 1.5\max\left[P_{sc}, \ P_{si}\right] \tag{4.19.53}$$

For reinforced and toroidal bellows, use $P_{si} = 0$ in the above equation.

b) The pressure testing of an expansion joint may be performed as a part of the vessel pressure test, provided the joint is accessible for inspection during pressure testing.

c) Any expansion joint restraining elements (see Figure 4.19.1 Sketch (b) and paragraph 4.19.3.1(c)) shall also be pressure tested in accordance with paragraph 4.1.6.2 and Part 8 as a part of the initial expansion joint pressure test, or as a part of the final vessel pressure test after installation of the joint.

4.19.12 Marking and Reports

a) The expansion joint Manufacturer, whether the vessel Manufacturer or a parts Manufacturer, shall have a valid ASME Code U2 Certificate of Authorization and shall complete the appropriate Data Report in accordance with Part 2.

b) The Manufacturer responsible for the expansion joint design shall include the following additional data and statements on the appropriate Data Report:

   1) Axial movement (+ and -), associated design life in cycles, and associated loading condition, if applicable;

   2) Spring rate; and

   3) That the expansion joint has been constructed to the rules of this paragraph.

c) A parts Manufacturer shall identify the vessel for which the expansion joint is intended on the Partial Data Report.

d) Markings shall not be stamped on the flexible elements of the expansion joint.

4.19.13 Specification Sheet for Expansion Joints

Specification sheets for expansion joints are provided in paragraph 4.19.17 for Metric and US customary units.

4.19.14 Nomenclature

A cross sectional metal area of one convolution.

A_c cross sectional metal area of all reinforcing collars for toroidal bellows.

A_f cross sectional metal area of one reinforcing fastener

A_r cross sectional metal area of one bellows reinforcing ring member.

α offset angle of the U-shape bellows side-wall.

B_1, B_2, B_3 stress and stiffness coefficients used for toroidal bellows.

C_p, C_f, C_d stress coefficients for U-shaped convolutions.

C_r convolution height factor for reinforced bellows.

C_1, C_2 coefficients used to determine the coefficients C_p, C_f, C_d

C_3 coefficient used to determine the coefficients B_1, B_2, B_3

C_r convolution height factor for reinforced bellows.

C_{wc} longitudinal weld joint efficiency for tangent collar.

C_{wr} longitudinal weld joint efficiency for reinforcing member.

D_b inside diameter of bellows convolution and end tangents.

D_c mean diameter of collar.

D_{eq} equivalent outside diameter.

D_m mean diameter of bellows convolution.

e_{eq} equivalent wall thickness.

E_b modulus of elasticity of bellows material at design temperature.

E_c modulus of elasticity of collar material at design temperature.

E_f modulus of elasticity of reinforcing fastener material at design temperature.

E_r modulus of elasticity of reinforcing ring member material at design temperature.

E_0 modulus of elasticity of bellows material at room temperature.

I_{xx} moment of inertia of one convolution cross section relative to the axis passing by the center of gravity and parallel to the axis of the bellows.

K_b bellows axial stiffness.

K_g fatigue strength reduction factor.

k factor considering the stiffening effect of the attachment weld and the end convolution on the pressure capacity of the end tangent

L_c bellows collar length.

L_t end tangent length.

L_f effective length of one reinforcing fastener.

L_w distance between toroidal bellows attachment welds.

N number of convolutions.

N_{alw} allowable number of fatigue cycles.

N_{spe} specified number of fatigue cycles.

n number of plies.

ν_b Poisson's ratio of bellows material.

P design pressure.

P_{sc} allowable internal design pressure to avoid column instability.

P_{si} allowable internal design pressure to avoid in-plane instability.

q convolution pitch as shown in Figure 4.19.1. If the convolutions show an off-set angle of the sidewalls in the neutral position, as shown in Figure 4.19.2, q is defined as the length between two consecutive convolutions when their sidewalls have been made parallel.

R ratio of the internal pressure force resisted by the bellows to the internal pressure force resisted by the reinforcement

r mean radius of toroidal bellows convolution.

r_{ic} crest convolution radius.

r_{ir} root convolution radius.

S allowable stress from Annex 3.A of bellows material at design temperature.

S_c allowable stress from Annex 3.A of collar material at design temperature.

S_f allowable stress from Annex 3.A of reinforcing fastener material at design temperature.

S_r allowable stress from Annex 3.A of reinforcing ring member material at design temperature.

S_t total stress range due to cyclic displacement.

S_y yield strength of the bellows material at the design temperature.

S_y^* effective yield strength of the bellows material at the design temperature.

S_1	circumferential membrane stress in bellows tangent, due to pressure P.
$S_1^{'}$	circumferential membrane stress in collar, due to pressure P.
S_2	circumferential membrane stress in bellows, due to pressure P.
$S_2^{'}$	circumferential membrane stress in reinforcing member, due to pressure P.
$S_2^{''}$	membrane stress in fastener, due to pressure P.
S_3	meridional membrane stress in bellows, due to pressure P.
S_4	meridional bending stress in bellows, due to pressure P.
S_5	meridional membrane stress in bellows, due to total equivalent axial displacement range Δq.
S_6	meridional bending stress in bellows, due to total equivalent axial displacement range Δq.
t	nominal thickness of one ply.
t_c	collar thickness.
t_p	thickness of one ply, corrected for thinning during forming.
U	usage factor.
w	convolution height.
Δq	total equivalent axial displacement range per convolution.

4.19.15 Tables

Table 4.19.1 – Stress Calculations And Acceptability Criteria For U-Shaped Unreinforced Bellows Subject to Internal Pressure

Stress Calculation
$S_1 = \dfrac{P\left(D_b + nt\right)^2 L_t E_b k}{2\left[nt\left(D_b + nt\right)L_t E_b + t_c D_c L_c E_c k\right]}$
$S_1' = \dfrac{P D_c^2 L_t E_c k}{2\left[nt\left(D_b + nt\right)L_t E_b + t_c D_c L_c E_c k\right]}$
$S_{2,E} = \dfrac{P\left[qD_m + L_t\left(D_b + nt\right)\right]}{2\left(A + nt_p L_t + t_c L_c\right)}$ *for end convolutions*
$S_{2,I} = \dfrac{PqD_m}{2A}$ *for intermediate convolutions*
$S_3 = \dfrac{Pw}{2nt_p}$
$S_4 = \dfrac{PC_p}{2n}\left(\dfrac{w}{t_p}\right)^2$
where
$k = \min\left[\left(\dfrac{L_t}{1.5\sqrt{D_b t}}\right),\ 1.0\right]$ $\quad t_P = t\sqrt{\dfrac{D_b}{D_m}}$
$A = \left[\left(\dfrac{\pi - 2}{2}\right)q + 2w\right]nt_p$
$D_c = D_b + 2\ nt + t_c$
$D_m = D_b + w + nt$
Acceptance Criteria
$S_1 \le S$
$S_1' \le C_{wc}\ S_c$
$S_{2,E} \le S$
$S_{2,I} \le S$
$S_3 + S_4 \le K_f S$
Notes:
1) $K_f = 3.0$ for as-formed bellows and $K_f = 1.5$ for annealed bellows.
2) C_p is evaluated using Tables 4.19.2.

Table 4.19.2 – Method To Determine Coefficient C_p

C_2	$C_1 \le 0.3$					
	α_0	α_1	α_2	α_3	α_4	α_5
0.2	1.001	-0.448	-1.244	1.932	-0.398	-0.291
0.4	0.999	-0.735	0.106	-0.585	1.787	-1.022
0.6	0.961	-1.146	3.023	-7.488	8.824	-3.634
0.8	0.955	-2.708	7.279	14.212	-104.242	133.333
1.0	0.95	-2.524	10.402	-93.848	423.636	-613.333
1.2	0.95	-2.296	1.63	16.03	-113.939	240
1.4	0.95	-2.477	7.823	-49.394	141.212	-106.667
1.6	0.95	-2.027	-5.264	48.303	-139.394	160
2.0	0.95	-2.073	-3.622	29.136	-49.394	13.333
2.5	0.95	-2.073	-3.622	29.136	-49.394	13.333
3.0	0.95	-2.073	-3.622	29.136	-49.394	13.333
3.5	0.95	-2.073	-3.622	29.136	-49.394	13.333
4.0	0.95	-2.073	-3.622	29.136	-49.394	13.333
C_2	$C_1 > 0.3$					
	α_0	α_1	α_2	α_3	α_4	α_5
0.2	1.001	-0.448	-1.244	1.932	-0.398	-0.291
0.4	0.999	-0.735	0.106	-0.585	1.787	-1.022
0.6	0.961	-1.146	3.023	-7.488	8.824	-3.634
0.8	0.622	1.685	-9.347	18.447	-15.991	5.119
1.0	0.201	2.317	-5.956	7.594	-4.945	1.299
1.2	0.598	-0.99	3.741	-6.453	5.107	-1.527
1.4	0.473	-0.029	-0.015	-0.03	0.016	0.016
1.6	0.477	-0.146	-0.018	0.037	0.097	-0.067
2.0	0.935	-3.613	9.456	-13.228	9.355	-2.613
2.5	1.575	-8.646	24.368	-35.239	25.313	-7.157
3.0	1.464	-7.098	17.875	-23.778	15.953	-4.245
3.5	1.495	-6.904	16.024	-19.6	12.069	-2.944
4.0	2.037	-11.037	28.276	-37.655	25.213	-6.716

Notes:

1) $C_1 = \dfrac{q}{2w}$ $\qquad 0 \le C_1 \le 1$

2) $C_2 = \dfrac{q}{2.2\sqrt{D_m t_p}}$ $\qquad 0.2 \le C_2 \le 4.0$

3) $C_p = \alpha_0 + \alpha_1 C_1 + \alpha_2 C_1^2 + \alpha_3 C_1^3 + \alpha_4 C_1^4 + \alpha_5 C_1^5$

4) A plot of C_p versus C_1 and C_2 is shown in Figure 4.19.11

Table 4.19.3 – Method To Determine Coefficient C_f

C_2	α_0	α_1	α_2	α_3	α_4	α_5
0.2	1.006	2.375	-3.977	8.297	-8.394	3.194
0.4	1.007	1.82	-1.818	2.981	-2.43	0.87
0.6	1.003	1.993	-5.055	12.896	-14.429	5.897
0.8	1.003	1.338	-1.717	1.908	0.02	-0.55
1.0	0.997	0.621	-0.907	2.429	-2.901	1.361
1.2	1	0.112	-1.41	3.483	-3.044	1.013
1.4	1	-0.285	-1.309	3.662	-3.467	1.191
1.6	1.001	-0.494	-1.879	4.959	-4.569	1.543
2.0	1.002	-1.061	-0.715	3.103	-3.016	0.99
2.5	1	-1.31	-0.829	4.116	-4.36	1.55
3.0	0.999	-1.521	-0.039	2.121	-2.215	0.77
3.5	0.998	-1.896	1.839	-2.047	1.852	-0.664
4.0	1	-2.007	1.62	-0.538	-0.261	0.249

Notes:

1) $C_1 = \dfrac{q}{2w}$ $\qquad 0 \le C_1 \le 1$

2) $C_2 = \dfrac{q}{2.2\sqrt{D_m t_p}}$ $\qquad 0.2 \le C_2 \le 4.0$

3) $C_f = \alpha_0 + \alpha_1 C_1 + \alpha_2 C_1^2 + \alpha_3 C_1^3 + \alpha_4 C_1^4 + \alpha_5 C_1^5$

4) A plot of C_f versus C_1 and C_2 is shown in Figure 4.19.12

Table 4.19.4 – Method To Determine Coefficient C_d

C_2	α_0	α_1	α_2	α_3	α_4	α_5
0.2	1	1.151	1.685	-4.414	4.564	-1.645
0.4	0.999	1.31	0.909	-2.407	2.273	-0.706
0.6	1.003	2.189	-3.192	5.928	-5.576	2.07
0.8	1.005	1.263	5.184	-13.929	13.828	-4.83
1.0	1.001	0.953	3.924	-8.773	10.44	-4.749
1.2	1.002	0.602	2.11	-3.625	5.166	-2.312
1.4	0.998	0.309	1.135	-1.04	1.296	-0.087
1.6	0.999	0.12	0.351	-0.178	0.942	-0.115
2.0	1	-0.133	-0.46	1.596	-1.521	0.877
2.5	1	-0.323	-1.118	3.73	-4.453	2.055
3.0	1	-0.545	-0.42	1.457	-1.561	0.71
3.5	1	-0.704	-0.179	0.946	-1.038	0.474
4.0	1.001	-0.955	0.577	-0.462	0.181	0.08

Notes:

1) $C_1 = \dfrac{q}{2w}$ $\qquad 0 \le C_1 \le 1$

2) $C_2 = \dfrac{q}{2.2\sqrt{D_m t_p}}$ $\qquad 0.2 \le C_2 \le 4.0$

3) $C_d = \alpha_0 + \alpha_1 C_1 + \alpha_2 C_1^2 + \alpha_3 C_1^3 + \alpha_4 C_1^4 + \alpha_5 C_1^5$

4) A plot of C_d versus C_1 and C_2 is shown in Figure 4.19.13

Table 4.19.5 – Allowable Number Of Cycles For U-Shaped Unreinforced Bellows

Stress in MPa	
Stress Range Criteria	**Allowable Number Of Cycles**
$K_g\left(E_o/E_b\right)S_t \geq 448\ MPa$	$N_{alw}=\left(\dfrac{35850}{K_g\left(E_o/E_b\right)S_t-264}\right)^2$
$257.2\ MPa < K_g\left(E_o/E_b\right)S_t < 448\ MPa$	$N_{alw}=\left(\dfrac{46200}{K_g\left(E_o/E_b\right)S_t-211}\right)^2$
$K_g\left(E_o/E_b\right)S_t \leq 257.2\ MPa$	$N_{alw}=10^6\ cycles$
Stress Range in psi	
Stress Range Criteria	**Allowable Number Of Cycles**
$K_g\left(E_o/E_b\right)S_t \geq 65000\ psi$	$N_{alw}=\left(\dfrac{5.2(10)^6}{K_g\left(E_o/E_b\right)S_t-38300}\right)^2$
$37300\ psi < K_g\left(E_o/E_b\right)S_t < 65000\ psi$	$N_{alw}=\left(\dfrac{6.7(10)^6}{K_g\left(E_o/E_b\right)S_t-30600}\right)^2$
$K_g\left(E_o/E_b\right)S_t \leq 37300\ psi$	$N_{alw}=10^6\ cycles$

Notes:

1) In the above equations, K_g is the fatigue strength reduction factor that accounts for geometrical stress concentration factors due to thickness variations, weld geometries, surface notches, and other surface or environmental conditions. The range for K_g is $1.0 \leq K_g \leq 4.0$ with its minimum value for smooth geometrical shapes and its maximum for 90 deg. welded corners and fillet welds. Fatigue strength reduction factors may be determined from theoretical, experimental, or photoelastic studies. A factor has already been included in the above equations for N to account for normal effects of size, environment, and surface finish. If the expansion bellows does not have circumferential welds and satisfies all of the design and examination requirements of this paragraph, a $K_g = 1.0$ may be used.

2) The allowable number of cycles given in this table includes a reasonable design margin (3 on cycles and 1.25 on stress) and represents the maximum number of cycles for the operating condition considered. Therefore an additional design margin should not be applied. An overly conservative estimate of cycles can necessitate a greater number of convolutions and result in a bellows more prone to instability.

Table 4.19.6 – Stress Calculations And Acceptability Criteria For U-Shaped Reinforced Bellows Subject To Internal Pressure

Stress Calculation	
Bellows	**Reinforcing Ring Member and Fastener**
$S_1 = \dfrac{P\left(D_b + nt\right)^2 L_t E_b k}{2\left[nt\left(D_b + nt\right)L_t E_b + t_c D_c L_c E_c k\right]}$ $S_1^{'} = \dfrac{P D_c^2 L_t E_c k}{2\left[nt\left(D_b + nt\right)L_t E_b + t_c D_c L_c E_c k\right]}$ $S_2 = \dfrac{P D_m q}{2A}\left(\dfrac{R}{R+1}\right)$ $S_3 = \dfrac{0.85\left(w - C_r\ q\right)P}{2nt_p}$ $S_4 = \dfrac{0.85 C_p P}{2n}\left(\dfrac{w - C_r\ q}{t_p}\right)^2$	$S_2^{'} = \dfrac{P D_m q}{2A_r}\left(\dfrac{1}{R_1 + 1}\right)$ *ring member* $S_2^{''} = \dfrac{P D_m q}{2A_f}\left(\dfrac{1}{R_2 + 1}\right)$ *fastener*
$R_1 = \dfrac{A\ E_b}{A_r\ E_r}$ $C_r = 0.3 - \left(\dfrac{100}{1048P^{1.5} + 320}\right)^2$ $P\ in\ MPa$ $R_2 = \dfrac{AE_b}{D_m}\left(\dfrac{L_f}{A_f E_f} + \dfrac{D_m}{A_r E_r}\right)$ $C_r = 0.3 - \left(\dfrac{100}{0.6P^{1.5} + 320}\right)^2$ $P\ in\ psig$	
Acceptance Criteria	
Bellows	**Reinforcing Ring Member and Fastener**
$S_1 \le S$ $S_1^{'} \le C_{wc}\, S_c$ $S_2 \le S$ $S_3 + S_4 \le K_f S$	$S_2^{'} \le C_{wr} S_r$ *ring member* $S_2^{''} \le S_f$ *fastener*

Notes:

1. k, A, D_c, D_m, t_p and K_f are evaluated using the equations in Table 4.19.1.
2. C_p is evaluated using Table 4.19.2.
3. $R = R_1$ for integral reinforcing ring members and $R = R_2$ for reinforcing ring members joined by fasteners.
4. In the case of reinforcing members which are made in sections, and joined by fasteners in tension, the equation for S_2 assumes that the structure used to retain the fastener does not bend so as to permit the reinforcing member to expand diametrically. In addition, the end reinforcing members must be restrained against the longitudinal annular pressure load of the bellows.
5. In the case of equalizing rings, the equation for $S_2^{'}$ provides only the simple membrane stress and does not include the bending stress caused by the eccentric fastener location. Elastic analysis and/or actual tests can determine these stresses.

Table 4.19.7 – Allowable Number Of Cycles For U-Shaped Reinforced Bellows

Stress in MPa	
Stress Range Criteria	**Allowable Number Of Cycles**
$K_g\left(E_o/E_b\right)S_t \geq 567\ MPa$	$N_{alw} = \left(\dfrac{45505}{K_g\left(E_o/E_b\right)S_t - 334}\right)^2$
$326.1\ MPa < K_g\left(E_o/E_b\right)S_t < 567\ MPa$	$N_{alw} = \left(\dfrac{58600}{K_g\left(E_o/E_b\right)S_t - 267.5}\right)^2$
$K_g\left(E_o/E_b\right)S_t \leq 326.1\ MPa$	$N_{alw} = 10^6\ cycles$
Stress Range in psi	
Stress Range Criteria	**Allowable Number Of Cycles**
$K_g\left(E_o/E_b\right)S_t \geq 82200\ psi$	$N_{alw} = \left(\dfrac{6.6(10)^6}{K_g\left(E_o/E_b\right)S_t - 48500}\right)^2$
$47300\ psi < K_g\left(E_o/E_b\right)S_t < 82200\ psi$	$N_{alw} = \left(\dfrac{8.5(10)^6}{K_g\left(E_o/E_b\right)S_t - 38800}\right)^2$
$K_g\left(E_o/E_b\right)S_t \leq 47300\ psi$	$N_{alw} = 10^6\ cycles$

Notes:

1) In the above equations, K_g is the fatigue strength reduction factor that accounts for geometrical stress concentration factors due to thickness variations, weld geometries, surface notches, and other surface or environmental conditions. The range for K_g is $1.0 \leq K_g \leq 4.0$ with its minimum value for smooth geometrical shapes and its maximum for 90 deg. welded corners and fillet welds. Fatigue strength reduction factors may be determined from theoretical, experimental, or photoelastic studies. A factor has already been included in the above equations for N to account for normal effects of size, environment, and surface finish. If the expansion bellows does not have circumferential welds and satisfies all of the design and examination requirements of this paragraph, $K_g = 1.0$ may be used.

2) The allowable number of cycles given in this table includes a reasonable design margin (3 on cycles and 1.25 on stress) and represents the maximum number of cycles for the operating condition considered. Therefore an additional design margin should not be applied. An overly conservative estimate of cycles can necessitate a greater number of convolutions and result in a bellows more prone to instability.

Table 4.19.8 – Stress Calculations And Acceptability Criteria For Toroidal Bellows Subject To Internal Pressure

Stress Calculation
$S_1 = \dfrac{P\left(D_b + nt\right)^2 L_w E_b}{2\left[nt\left(D_b + nt\right) L_w E_b + D_c E_c A_c\right]}$ $S_1^{'} = \dfrac{P D_c^2 L_w E_c}{2\left[nt\left(D_b + nt\right) L_w E_b + D_c E_c A_c\right]}$ $S_2 = \dfrac{Pr}{2nt_p}$ $S_3 = \dfrac{Pr}{nt_p}\left(\dfrac{D_m - r}{D_m - 2r}\right)$ where $D_c = D_b + 2\ nt + t_c$
Acceptance Criteria
$S_1 \le S$ $S_1^{'} \le C_{wc} S_c$ $S_2 \le S$ $S_3 \le S$

Table 4.19.9 – Stress And Axial Stiffness Coefficients For Toroidal Bellows

C_3	B_1	B_2	B_3
0	1.0	1.0	1.0
1	1.1	1.0	1.1
2	1.4	1.0	1.3
3	2.0	1.0	1.5
4	2.8	1.0	1.9
5	3.6	1.0	2.3
6	4.6	1.1	2.8
7	5.7	1.2	3.3
8	6.8	1.4	3.8
9	8.0	1.5	4.4
10	9.2	1.6	4.9
11	10.6	1.7	5.4
12	12.0	1.8	5.9
13	13.2	2.0	6.4
14	14.7	2.1	6.9
15	16.0	2.2	7.4
16	17.4	2.3	7.9
17	18.9	2.4	8.5
18	20.3	2.6	9.0
19	21.9	2.7	9.5
20	23.3	2.8	10.0

Notes: Equations for B_1, B_2 and B_3 are shown below.

$$B_1 = \frac{1.00404 + 0.028725C_3 + 0.18961C_3^2 - 0.00058626C_3^3}{1 + 0.14069C_3 - 0.0052319C_3^2 + 0.00029867C_3^3 - 6.2088(10)^{-6}C_3^4}$$

$$B_2 = 1.0 \qquad for \qquad C_3 \le 5$$

$$B_2 = \frac{0.049198 - 0.77774C_3 - 0.13013C_3^2 + 0.080371C_3^3}{1 - 2.81257C_3 + 0.63815C_3^2 + 0.0006405C_3^3} \qquad for \qquad C_3 > 5$$

$$B_3 = \frac{0.99916 - 0.091665C_3 + 0.040635C_3^2 - 0.0038483C_3^3 + 0.00013392C_3^4}{1 - 0.1527C_3 + 0.013446C_3^2 - 0.00062724C_3^3 + 1.4374(10)^{-5}C_3^4}$$

where,

$$C_3 = \frac{6.61r^2}{D_m t_p}$$

Table 4.19.10 – Allowable Number Of Cycles For Toroidal Bellows

Stress in MPa	
Stress Range Criteria	**Allowable Number Of Cycles**
$K_g\left(E_o/E_b\right)S_t \geq 448\ MPa$	$N_{alw} = \left(\dfrac{35850}{K_g\left(E_o/E_b\right)S_t - 264}\right)^2$
$257.2\,MPa < K_g\left(E_o/E_b\right)S_t < 448\ MPa$	$N_{alw} = \left(\dfrac{46200}{K_g\left(E_o/E_b\right)S_t - 211}\right)^2$
$K_g\left(E_o/E_b\right)S_t \leq 257.2\ MPa$	$N_{alw} = 10^6\ cycles$
Stress Range in psi	
Stress Range Criteria	**Allowable Number Of Cycles**
$K_g\left(E_o/E_b\right)S_t \geq 65000\ psi$	$N_{alw} = \left(\dfrac{5.2(10)^6}{K_g\left(E_o/E_b\right)S_t - 38300}\right)^2$
$37300\ psi < K_g\left(E_o/E_b\right)S_t < 65000\ psi$	$N_{alw} = \left(\dfrac{6.7(10)^6}{K_g\left(E_o/E_b\right)S_t - 30600}\right)^2$
$K_g\left(E_o/E_b\right)S_t \leq 37300\ psi$	$N_{alw} = 10^6\ cycles$

Notes:

1) In the above equations, K_g is the fatigue strength reduction factor that accounts for geometrical stress concentration factors due to thickness variations, weld geometries, surface notches, and other surface or environmental conditions. The range for K_g is $1.0 \leq K_g \leq 4.0$ with its minimum value for smooth geometrical shapes and its maximum for 90 deg. welded corners and fillet welds. Fatigue strength reduction factors may be determined from theoretical, experimental, or photoelastic studies. A factor has already been included in the above equations for N to account for normal effects of size, environment, and surface finish. If the expansion bellows does not have circumferential welds and satisfies all of the design and examination requirements of this paragraph, $K_g = 1.0$ may be used.

2) The allowable number of cycles given in this table includes a reasonable design margin (3 on cycles and 1.25 on stress) and represents the maximum number of cycles for the operating condition considered. Therefore an additional design margin should not be applied. An overly conservative estimate of cycles can necessitate a greater number of convolutions and result in a bellows more prone to instability.

4.19.16 Figures

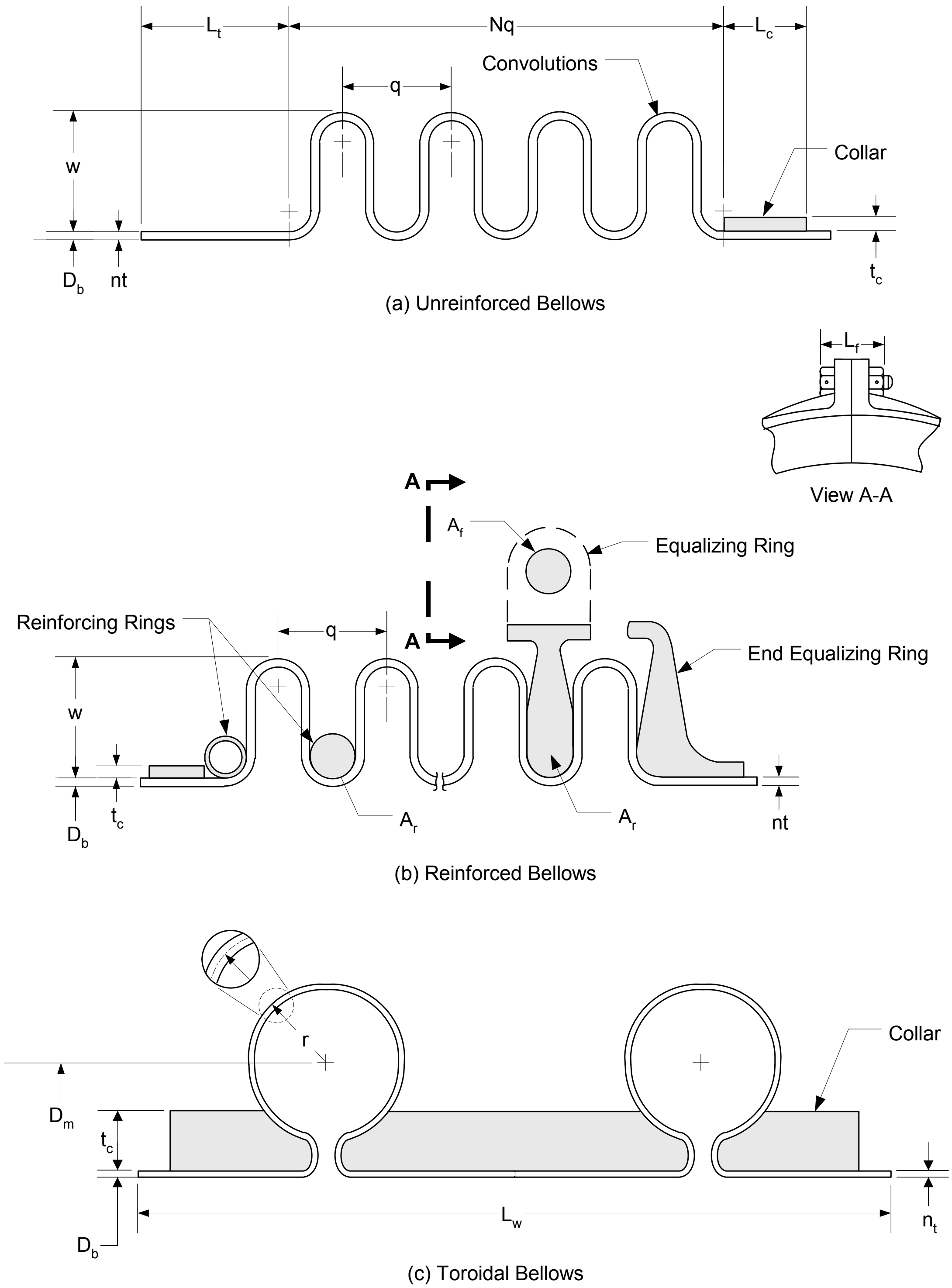


Figure 4.19.1 – Typical Bellows Expansion Joints

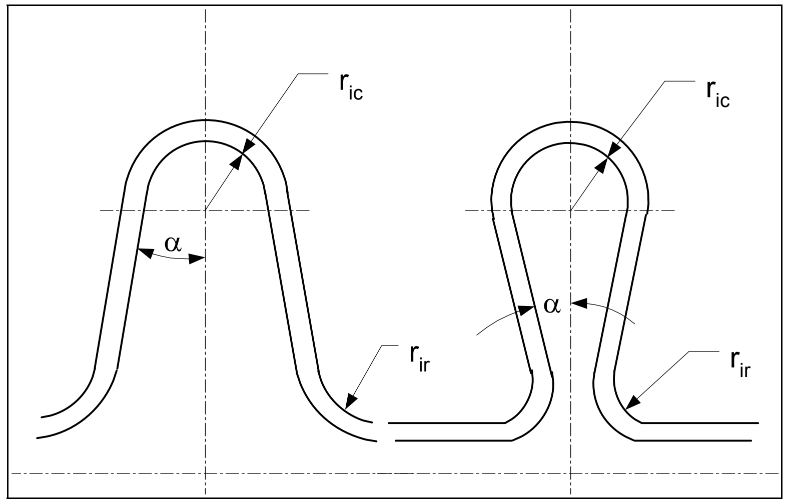


Figure 4.19.2 – Possible Convolution Profile in Neutral Position

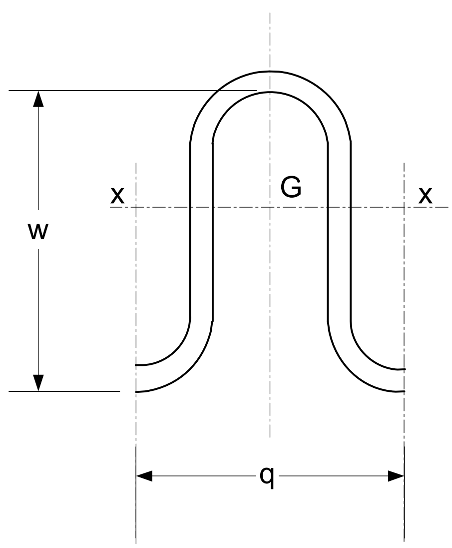


Figure 4.19.3 – Dimensions to Determine I_{xx}

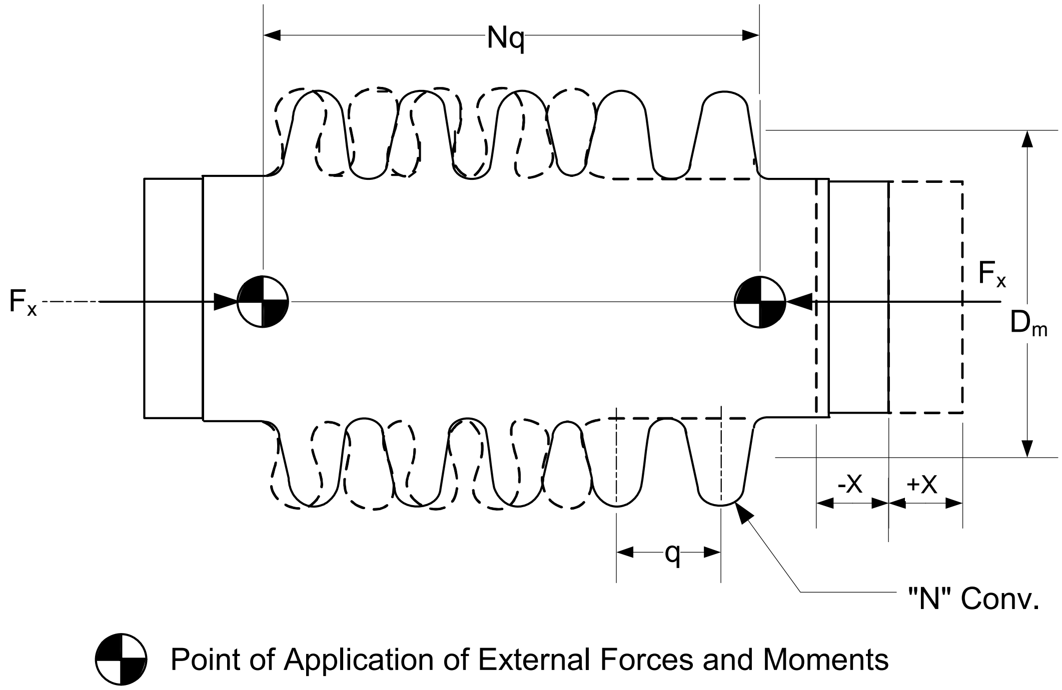


Figure 4.19.4 – Bellows Subject to an Axial Displacement x

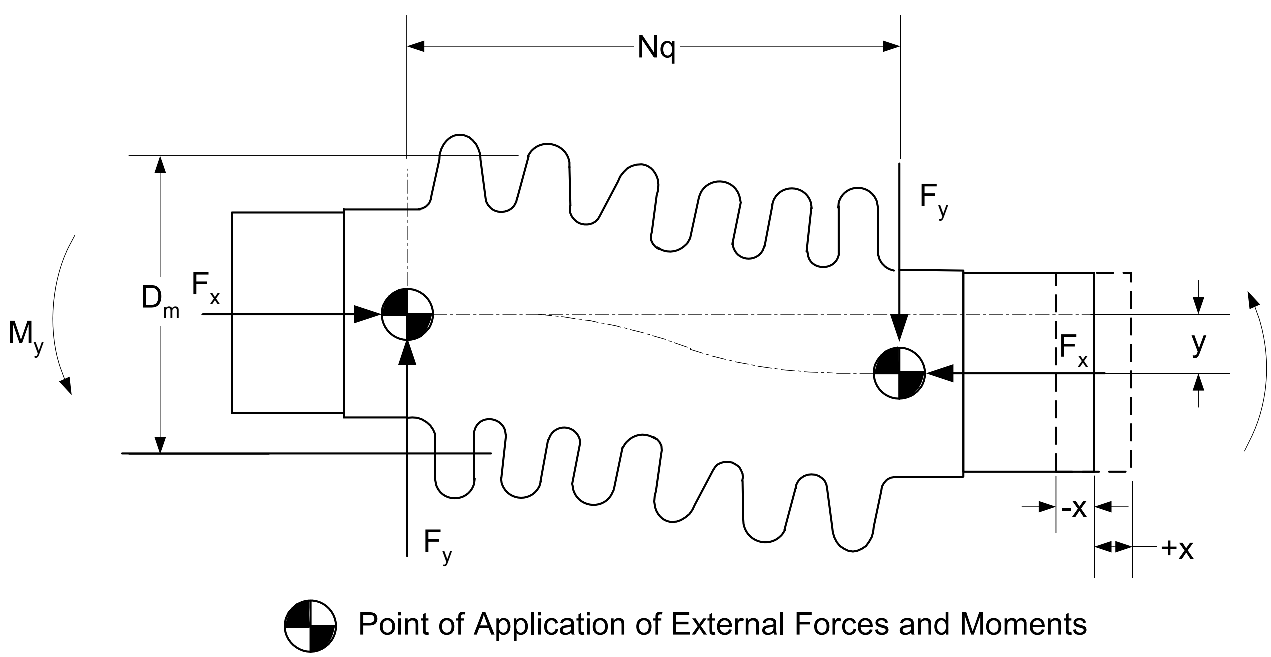


Figure 4.19.5 – Bellows Subject to a Lateral Displacement y

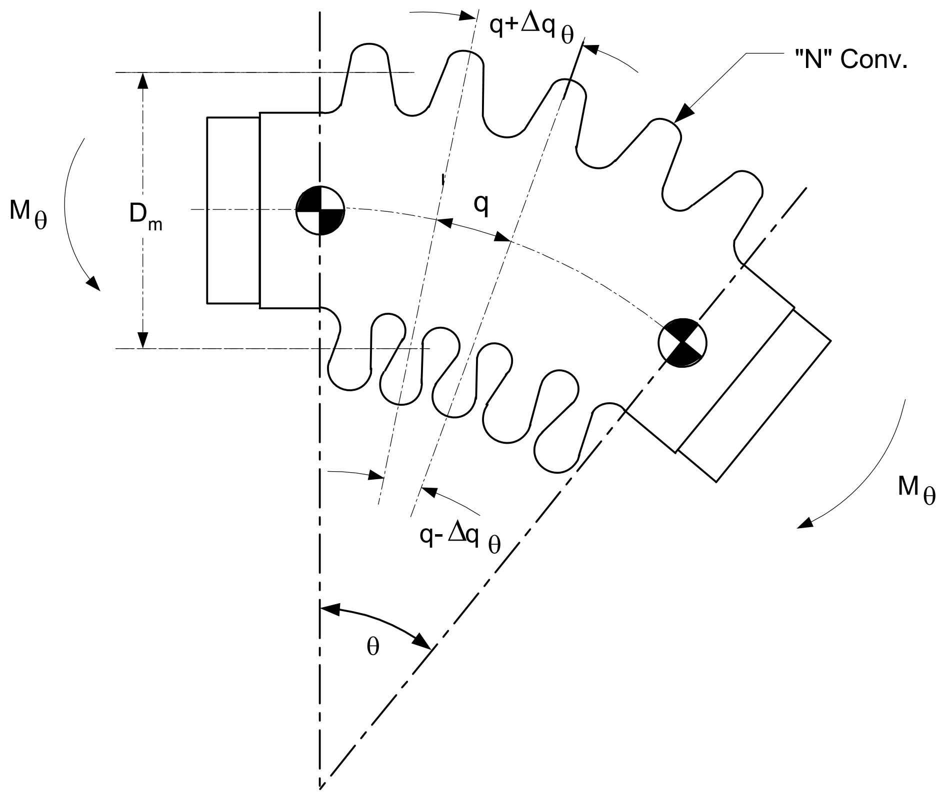


Figure 4.19.6 – Bellows Subjected to an Angular Rotation θ

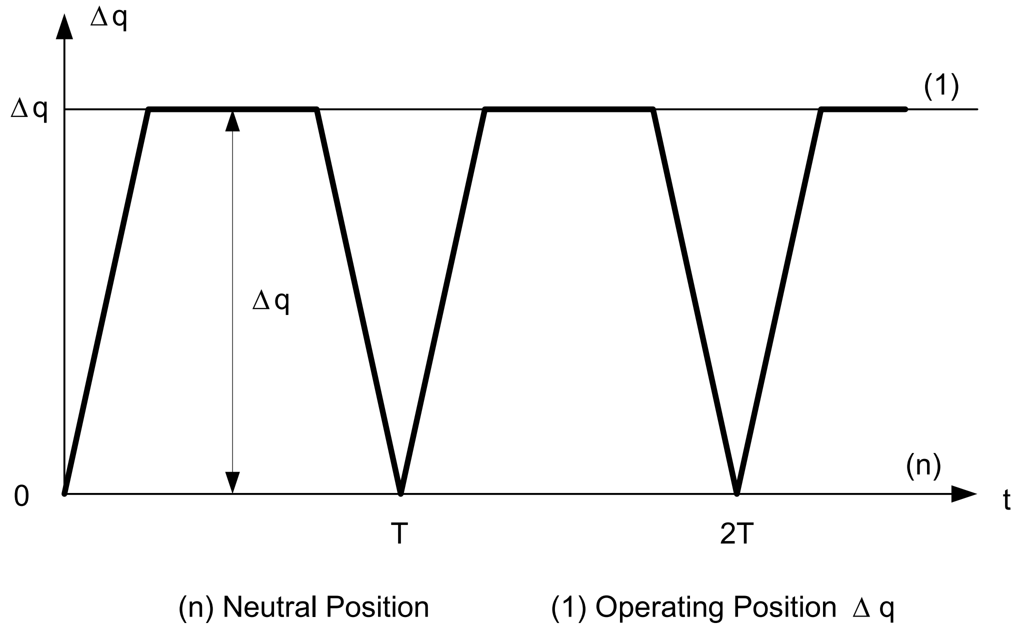


Figure 4.19.7 – Cyclic Displacements

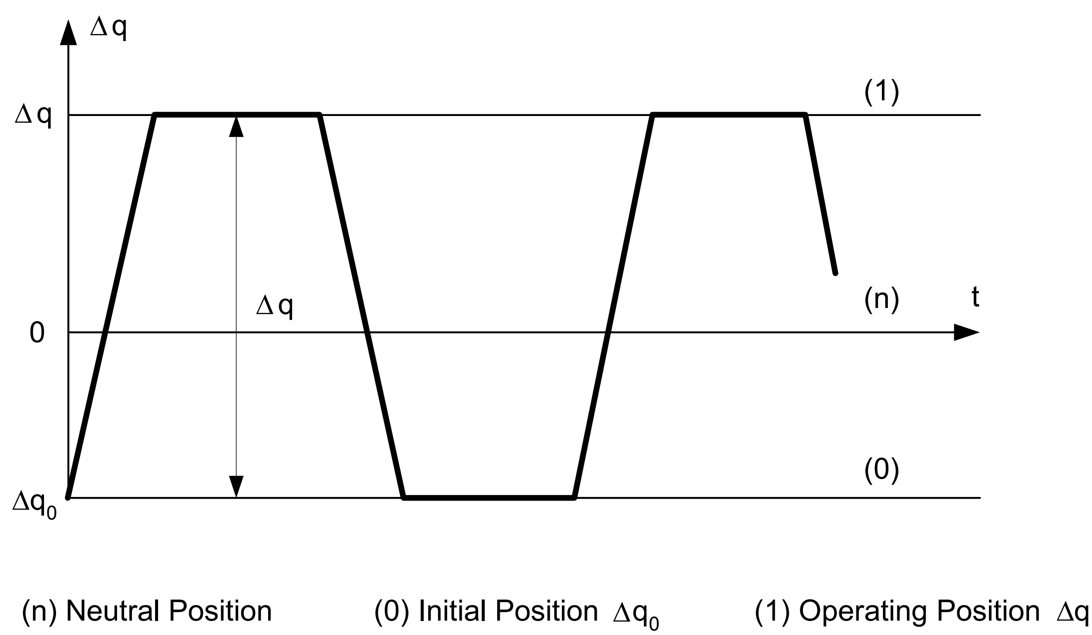
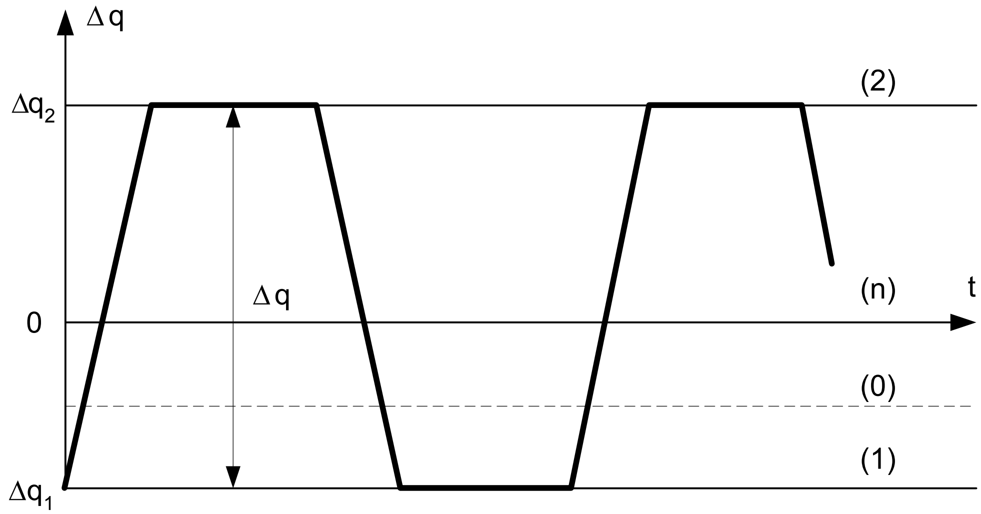


Figure 4.19.8 – Cyclic Displacements



(0) Initial Position 0
(n) Neutral Position
(1) Operating Position 1
(2) Operating Position 2

Figure 4.19.9 – Cyclic Displacements

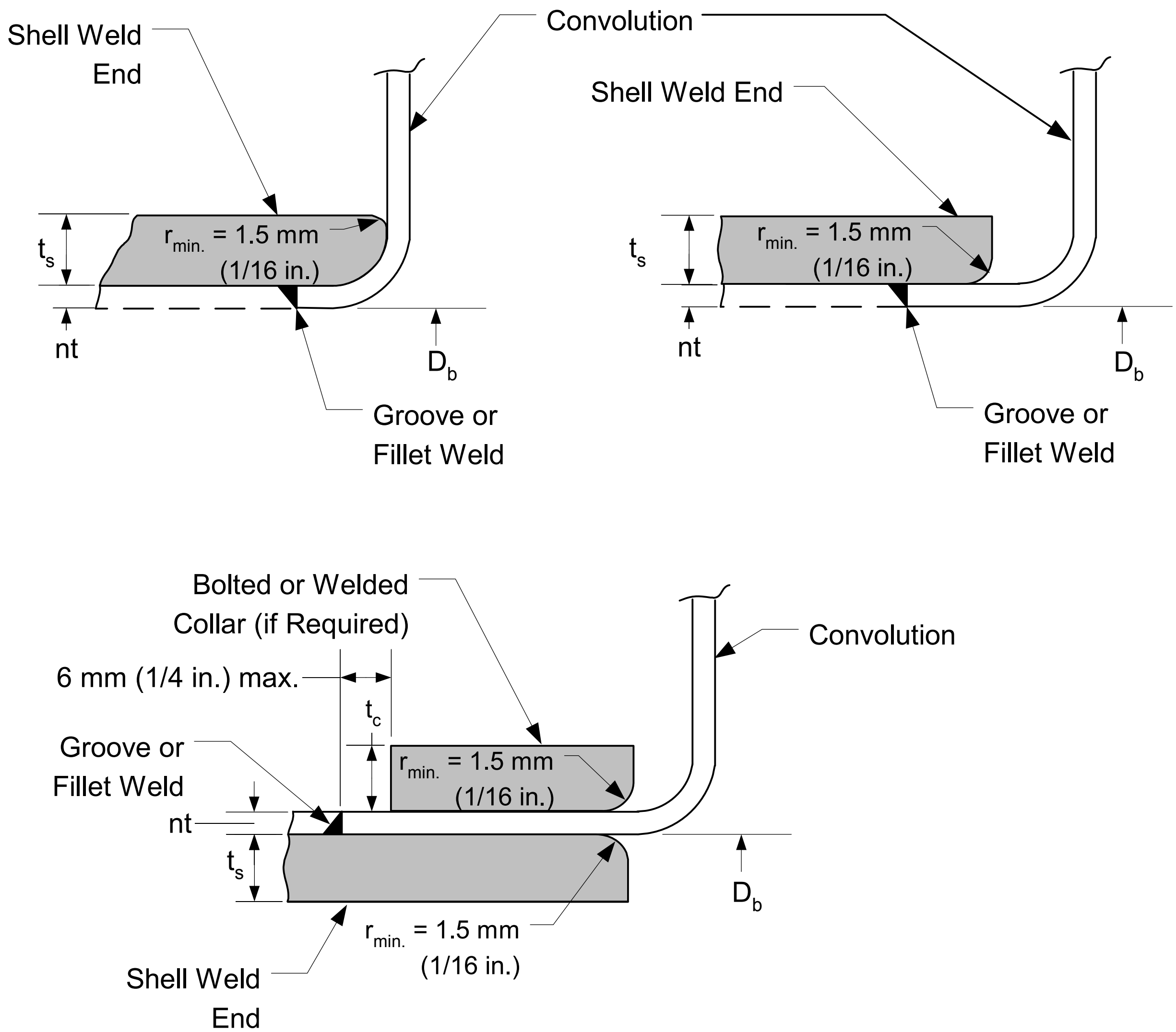


Figure 4.19.10 – Some Typical Expansion Bellows Attachment Welds

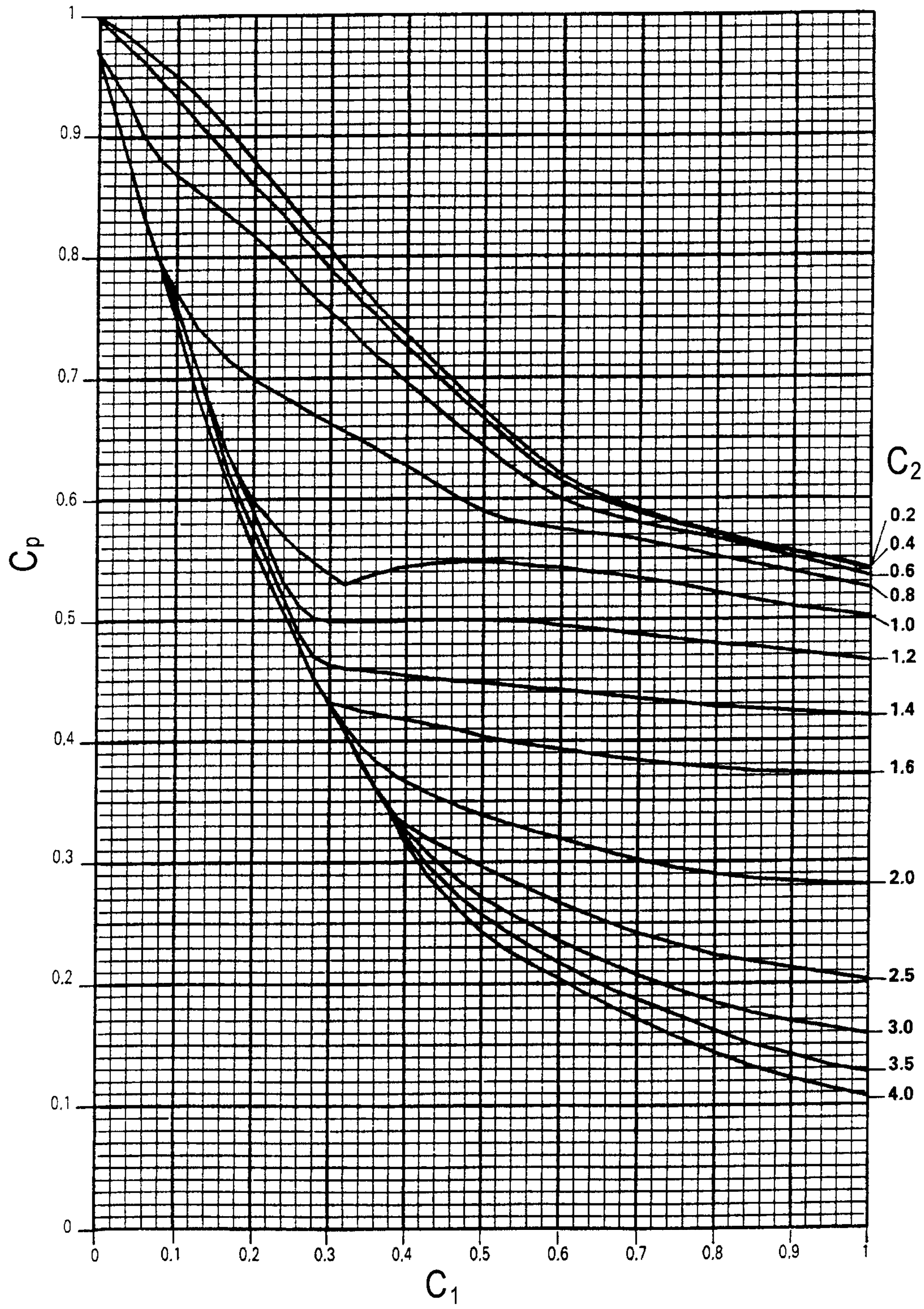


Figure 4.19.11 – C_p **Versus** C_1 **and** C_2

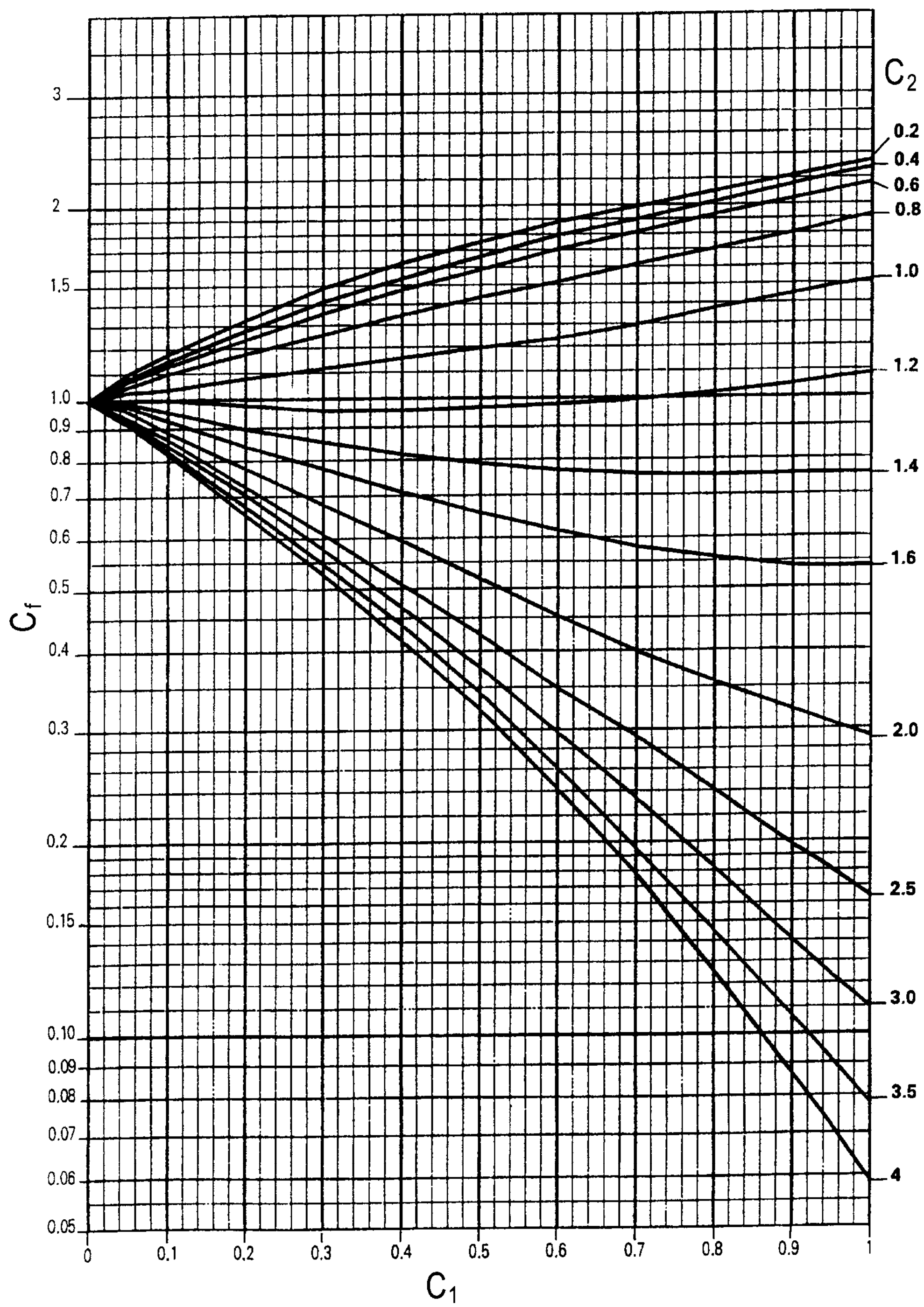


Figure 4.19.12 – C_f Versus C_1 and C_2

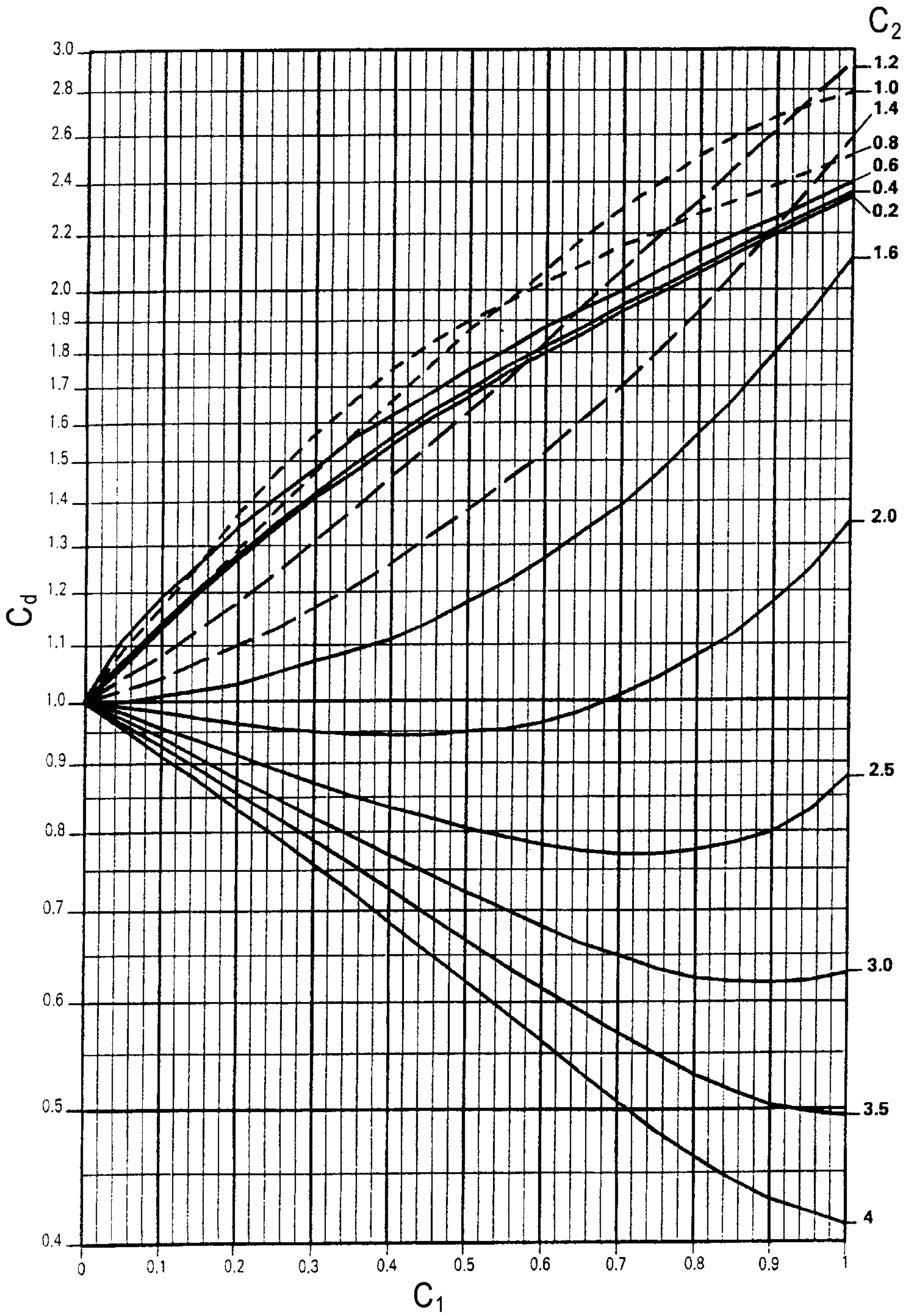


Figure 4.19.13 – C_d Versus C_1 and C_2

4.19.17 Specification Sheets

Specification Sheet For ASME Section VIII, Division 2 Expansion Joints – Metric Units

Date:_______/_________/__________ Applicable ASME Code Edition:____________________

1. Item Number:____________________ Vessel Manufacturer:______________________________
2. Drawing/Tag/Serial/Job Number:__________________ Vessel Owner:___________________________________
3. Quantity:____________________ Installation Location:______________________________
4. Size: _________ OD __________ID mm Expansion Joint Overall Length:_________________ mm

5. Internal Pressure: Design _________ MPa
6. External Pressure: Design _________ MPa
7. Vessel Manufacturer Hydrotest Pressure Internal _________ MPa External _________ MPa

8. Temperature	Design _________ ^{0}C	Operating _________ ^{0}C	Upset _________ ^{0}C
9. Vessel rating	MAWP _________ MPa	MDMT _________ ^{0}C	Installed Position: Horz. Vert.

10. Design Movements: Axial Compression: (–)______mm Axial Extension: (+)______mm Lateral:______ mm Angular:______ deg
11. Specified number of Cycles: ______________

12. Shell Material: ______________ Bellows Material: ______________
13. Shell thickness ______________ mm Shell Corrosion Allowance: Internal:________ mm External:__________ mm
14. Shell Radiography: Spot / Full
15. End Preparation: Square Cut Outside Bevel Inside Bevel Double Bevel (Describe in Line 23 if special)
16. Heat Exchanger Tube Length Between Inner Tubesheet Faces: ________ mm

17. Maximum Bellows Spring Rate: No Yes – _________________________ N/mm
18. Internal Liner: No Yes – Material_______________________________
19. Drain Holes in Liner: No Yes – Quantity/Size:_________________________
20. Liner Flush with Shell ID: No Yes – Telescoping Liners? No Yes
21. External Cover: No Yes – Material:______________________________
22. Pre-Production Approvals Required: No Yes – Drawings / Bellows Calculations / Weld Procedures

23. Additional Recommendations: (i.e. bellows pre-set, ultrasonic examination, etc.)

Temporary shipping bars are required to maintain assembly length during shipping and vessel fabrication only, and ARE NOT to be used during vessel hydrotest for expansion joint pressure restraint (see paragraph 4.19.3.1.c and 4.19.3.1.d.)

Specification Sheet For ASME Section VIII, Division 2 Expansion Joints – US Customary Units

Date:_______/_________/__________ Applicable ASME Code Edition:____________________

1. Item Number:____________________ Vessel Manufacturer:_____________________________

2. Drawing/Tag/Serial/Job Number:__________________ Vessel Owner:__________________________________

3. Quantity:____________________ Installation Location:_____________________________

4. Size: _________ OD __________ID in. Expansion Joint Overall Length:_________________ in.

5. Internal Pressure: Design _________ psig

6. External Pressure: Design _________ psig

7. Vessel Manufacturer Hydrotest Pressure Internal _________ psig External _________ psig

8. Temperature	Design _________ ^{0}F	Operating _________ ^{0}F	Upset _________ ^{0}F
9. Vessel rating	MAWP _________ psig	MDMT _________ ^{0}F	Installed Position: Horz. Vert.

10. Design Movements: Axial Compression: (–)______in. Axial Extension: (+)______in. Lateral:______ in. Angular:______ deg

11. Specified number of Cycles: ______________

12. Shell Material: ______________ Bellows Material: ______________

13. Shell thickness ______________ in. Shell Corrosion Allowance: Internal: ________ in. External:__________ in.

14. Shell Radiography: Spot / Full

15. End Preparation: Square Cut Outside Bevel Inside Bevel Double Bevel (Describe in Line 23 if special)

16. Heat Exchanger Tube Length Between Inner Tubesheet Faces: ________ in.

17. Maximum Bellows Spring Rate:	No	Yes – _________________________ lbs/in
18. Internal Liner:	No	Yes – Material_______________________________
19. Drain Holes in Liner:	No	Yes – Quantity/Size:_________________________
20. Liner Flush with Shell ID:	No	Yes – Telescoping Liners? No Yes
21. External Cover:	No	Yes – Material:______________________________
22. Pre-Production Approvals Required:	No	Yes – Drawings / Bellows Calculations / Weld Procedures

23. Additional Recommendations: (i.e. bellows pre-set, ultrasonic examination, etc.)

Temporary shipping bars are required to maintain assembly length during shipping and vessel fabrication only, and ARE NOT to be used during vessel hydrotest for expansion joint pressure restraint (see paragraph 4.19.3.1.c and 4.19.3.1.d.)

ANNEX 4.A

CURRENTLY NOT USED

ANNEX 4.B

GUIDE FOR THE DESIGN AND OPERATION OF QUICK-ACTUATING (QUICK-OPENING) CLOSURES

(INFORMATIVE)

4.B.1 Introduction

4.B.1.1 This Annex provides guidance in the form of recommendations for the installation, operation, and maintenance of quick-actuating closures. This guidance is primarily for the use of the Owner and the user. The safety of the quick-actuating closure is the responsibility of the user. This includes the requirement for the user to provide training for all operating personnel, follow safety procedures, periodically inspect the closure, provide scheduled maintenance, and have all necessary repairs made in a timely fashion.

4.B.1.2 This Annex also contains guidance for use by the Designer. The rules specific to the design and construction of quick-actuating closures are found in paragraph 4.8 of this division.

4.B.1.3 The Manufacturer should supply the Owner a copy(s) of the Installation, Operational, and Maintenance Manual for the quick-actuating closure which should, as a minimum, address the requirements described in this Annex. The Owner should supply a copy of the Installation, Operational, and Maintenance Manual to the user.

4.B.2 Responsibilities

4.B.2.1 It is the responsibility of the user to ensure that the sensing and safety devices and the equipment specified by the Manufacturer are properly installed before initial operation, and maintained during subsequent operation. Provision of written operation and maintenance procedures and training of personnel are also the responsibility of the Owner or user.

4.B.2.2 The user must not remove any devices furnished or specified by the Manufacturer of the vessel, and any repairs or replacements must be the same as, or equal to, the original equipment furnished or specified by the Manufacturer.

4.B.2.3 The rules of this Annex do not require these devices to be supplied by the Manufacturer of the vessel or of the quick-actuating closure.

4.B.3 Design

4.B.3.1 Code rules cannot be written to address each specific design; therefore, engineering judgment exercised by a qualified designer with the necessary experience is required to achieve a safe design. Because of the multiple requirements imposed on the design, it should be prepared by a designer with suitable experience and training in the design of quick-actuating closures.

4.B.3.2 The design must be safe, reliable, and allow for quick and safe opening and closing. Therefore, sensing and safety devices and equipment are integral and vitally important parts of the closure, and are to be furnished or specified by the Manufacturer of the vessel or quick-actuating closure. These devices must never be removed by the user.

4.B.3.3 It should be noted that there is a higher likelihood of personnel being close to the vessel and the closure when accidents during opening occur, especially those due to violations of operating procedures. An example is attempting to pry open the closure when they believe the vessel has been depressurized and it may not be.

4.B.3.4 The passive safety features described below can help to protect against such actions, but most can still be subverted. Protection against subversion of safety features is covered under Inspection, Training, and Administrative Controls.

4.B.3.5 Structural Elements in the vessel and the closure require design margins. However, it is also important to provide the suggested features listed below, for erroneous opening.

a) Passive Actuation – A passively actuated safety feature or device does not require the operator to take any action to provide safety. An example is a pressure relief valve in a vessel or a pressure-actuated locking device in a quick-actuating closure.

b) Redundancy – A redundant safety feature or device is one of two or more features or devices that perform the same safety function. Two pressure-actuated locking devices in parallel are an example application to quick-actuating closures. Another example is two or more independent holding elements, the failure of one of which does not reduce the capability to withstand pressure loadings below an acceptable level.

c) Fail-Safe Behavior – If a device or element fails, it should fail in a safe mode. An example applicable to quick-actuating closures is a normally-closed electrical interlock that stays locked if power fails.

d) Multiple Lines of Defense – This can consist of any combination of two or more items from the list above. They should consist, at the very least, of warnings or alarms.

4.B.3.6 Pressure controls and sensors that operate well at 350kPa or 700kPa (50psi or 100psi) or at much greater pressure do not operate well at very low pressure. For example, they may not sense a small, static head of hot fluid. Certain accidents can occur because of the release of hot fluid under static head alone, or under very low pressure. To protect against such accidents, separate controls and sensors may be used to maintain operating pressure on the one hand, and others may be required to prevent inappropriate opening at low pressures.

4.B.3.7 It may be necessary or desirable to utilize electrical or electronic devices and interlocks. If these are used, careful detailed installation, operating, and maintenance instructions (see following) shall be required.

4.B.3.8 The effects of repetitive loading shall be considered. There are two phenomena that are of major concern. The first is the wear produced by repetitive actuation of the mechanism. This can generally be mitigated by routine maintenance. The second is fatigue damage produced in the vessel or in the closure by repetitive actuation of the mechanism or by repetitive pressurization and depressurization.

4.B.3.9 The code does not provide explicit guidance for the evaluation or mitigation of wear. As well as proper maintenance, the selection of suitable materials for mating wear surfaces and control of contact stresses is necessary during the design process to properly control wear.

4.B.4 Installation

4.B.4.1 The manufacturer shall provide clear instructions for the installation of the quick-actuating closure itself and any adjustments that are necessary in the field. An example is adjustment of wedges or clamps.

4.B.4.2 Instructions, preferably including schematics and drawings, shall be provided for the installation, adjustment, and checkout of interlocks and warning devices.

4.B.4.3 Maintenance

4.B.4.4 Vessels with quick-actuating closures are commonly installed in industrial environments subject to dirt, moisture, abrasive materials, etc. These environmental factors are detrimental to safe and reliable operation of mechanical, electrical and electronic sensors and safety devices. Therefore, the user should establish a suitable cleaning and maintenance interval, and a means to verify that the equipment has been properly cleaned and maintained.

4.B.4.5 Accidents have occurred because gaskets have stuck, and have released suddenly when pried open. Many soft gaskets (60-70 Shore A Scale) have a combined shelf life and operating life of as little as six months. Aging can change the properties of the gasket material and change the gasket dimensions, impeding its proper function.

4.B.5 Inspection

4.B.5.1 It is recommended that the user inspect the completed installation including the pressure gauges before it is permitted to operate. Records of this inspection should be retained.

4.B.5.2 It is recommended that the user establish and document a periodic in-service inspection program, and that this program be followed and the results documented.

4.B.6 Training

4.B.6.1 Many accidents involving quick-actuating closures have occurred because the operators have been unfamiliar with the equipment or its safety features. The greater safety inherent in current designs has sometimes been produced by the use of sophisticated mechanical, electrical or electronic control devices. In order to make these features produce the maximum safety, personnel should be properly trained in their operation and maintenance.

4.B.6.2 Note that accidents may occur because hot fluid remains present in the vessel at atmospheric pressure of 15 kPa to 20 kPa (2 psig to 3 psig). If the vessel is forced open while under this pressure, then injuries may occur. Such specific accident sources should be guarded against by training and by administrative procedures. It is important that sound written operating procedures, understandable by the operating personnel and multi-lingual if necessary, exist for the quick-actuating closure, and that the operators be trained in the proper use of all interlocks, sensing devices, and manual closure mechanisms.

4.B.6.3 Provision of written operation and maintenance procedures and training of personnel are the responsibility of the user.

4.B.6.4 As part of the training program, testing should be performed to ensure that the trainee understands the material he or she is trained in. Records should be retained by the user.

4.B.7 Administrative Controls

The user should provide administrative controls covering training, cleanliness, operation, periodic inspection, and maintenance of equipment with quick-actuating closures. Records should be retained by the user.

ANNEX 4.C

BASIS FOR ESTABLISHING ALLOWABLE LOADS FOR TUBE-TO-TUBESHEET JOINTS

(NORMATIVE)

4.C.1 General

4.C.1.1 This Annex provides a basis for establishing allowable tube-to-tubesheet joint loads, except for the following.

a) Tube-to-tubesheet joints having full-strength welds as defined in accordance with paragraph 4.18.10.2.a shall be designed in accordance with paragraph 4.18.10.3 and do not require shear load testing.

b) Tube-to-tubesheet joints having partial-strength welds as defined in accordance with paragraph 4.18.10.2.b and designed in accordance with paragraph 4.18.10.4 do not require shear load testing.

4.C.1.2 The rules of this Annex are not intended to apply to U-tube construction.

4.C.1.3 Tubes used in the construction of heat exchangers or similar apparatus may be considered to act as stays which support or contribute to the strength of the tubesheets in which they are engaged. Tube-to-tubesheet joints shall be capable of transferring the applied tube loads. The design of tube-to-tubesheet joints depends on the type of joint, degree of examination, and shear load tests, if performed. Some acceptable geometries and combinations of welded and mechanical joints are described in Table 4.C.1. Some acceptable types of welded joints are illustrated in Figure 4.C.1.

a) Geometries, including variations in tube pitch, fastening methods, and combinations of fastening methods, not described or shown, may be used provided qualification tests have been conducted and applied in compliance with the procedures in paragraphs 4.C.3 and 4.C.4.

b) Materials for welded tube-to-tubesheet joints which do not meet the requirements of Part 6, but in all other respects meet the requirements of this Division, may be used if qualification tests of the tube-to-tubesheet joint have been conducted and applied in compliance with the procedures in paragraphs 4.C.3 and 4.C.4.

4.C.1.4 Some combinations of tube and tubesheet materials, when welded, result in welded joints having lower ductility than required in the material specifications. Appropriate tube-to-tubesheet joint geometry, welding method, and/or heat treatment shall be used with these materials to minimize this effect.

4.C.1.5 In the selection of joint type, consideration shall be given to the mean metal temperature of the joint at operating temperatures and differential thermal expansion of the tube and tubesheet which may affect the joint integrity. The following provisions apply for establishing maximum operating temperature for tube-to-tubesheet joints.

a) Tube-to-tubesheet joints made by welding shall be limited to the maximum temperature for which there are allowable stresses for the tube or tubesheet material in Annex 3.A.

b) Tube-to-tubesheet joints that depend on friction between the tube and the tube hole such as in Joint Types i, j, and k as shown in Table.4.C.1, shall be limited to temperatures as determined by the following.

   1) The operating temperature of the tube-to-tubesheet joint shall be within the tube and tubesheet time independent properties as provided in Annex 3.A.

2) The maximum operating temperature is based on the interface pressure that exists between the tube and tubesheet. The maximum operating temperature is limited such that the interface pressure due to expanding the tube at joint fabrication plus the interface pressure due to differential thermal expansion, $\left(P_o + P_T\right)$, does not exceed 58% of the smaller of the tube or tubesheet yield strength in Annex 3.D at the operating temperature. If the tube or tubesheet yield strength is not listed in Annex 3.D, the operating temperature limit shall be determined as described in 4.C.1.5.b.4. The interface pressure due to expanding the tube at fabrication or the interface pressure due to differential thermal expansion may be determined analytically or experimentally.

3) Due to differential thermal expansion, the tube may expand less than the tubesheet. For this condition, the interfacial pressure P_T is a negative number.

4) When the maximum temperature is not determined by 4.C.1.5.b.2, or the tube expands less than or equal to the tubesheet, joint acceptability shall be determined by shear load tests described in 4.C.3. Two sets of specimens shall be tested. The first set shall be tested at the proposed operating temperature. The second set shall be tested at room temperature after heat soaking at the proposed operating temperature for 24 hours. The proposed operating temperature is acceptable if the provisions of paragraph 4.C.5 are satisfied.

4.C.1.6 The Manufacturer shall prepare written procedures for joints which are expanded (whether welded and expanded or expanded only) for joint strength. The Manufacturer shall establish the variables that affect joint repeatability in these procedures. The procedures shall provide detailed descriptions or sketches of enhancements, such as grooves, serrations, threads, and coarse machining profiles. The Manufacturer shall make these written procedures available to the Authorized Inspector.

4.C.2 Maximum Axial Loads

The maximum allowable axial load in either direction on tube-to-tubesheet joints shall be determined in accordance with the following:

$$L_{\max} = A_t S_a f_r \qquad \textit{Joint Types } a,b,b\text{-}1,e \tag{4.C.1}$$

$$L_{\max} = A_t S_a f_e f_r f_y \qquad \textit{Joint Types } f,g,h \tag{4.C.2}$$

$$L_{\max} = A_t S_a f_e f_r f_y f_T \qquad \textit{Joint Types } i,j,k \tag{4.C.3}$$

where

$$A_t = \pi\left(d_o - t\right)t \tag{4.C.4}$$

$$S_a = kS \tag{4.C.5}$$

$$f_T = \frac{P_o + P_T}{P_o} \tag{4.C.6}$$

4.C.3 Shear Load Test

4.C.3.1 Flaws in the specimen may affect results. If any test specimen develops flaws, the retest provisions of paragraph 4.C.3.11 shall govern.

4.C.3.2 If any test specimen fails because of mechanical reasons, such as failure of testing equipment or improper specimen preparation, it may be discarded and another specimen taken from the same heat.

4.C.3.3 The shear load test subjects a full-size specimen of the tube joint under examination to a measured load sufficient to cause failure. In general, the testing equipment and methods are given in the Methods of Tension Testing of Metallic Materials (ASTM E 8). Additional fixtures for shear load testing of tube-to-tubesheet joints are shown in Figure 4.C.2.

4.C.3.4 The test block simulating the tubesheet may be circular, square or rectangular in shape, essentially in general conformity with the tube pitch geometry. The test assembly shall consist of an array of tubes such that the tube to be tested is in the geometric center of the array and completely surrounded by at least one row of adjacent tubes. The test block shall extend a distance of at least one tubesheet ligament beyond the edge of the peripheral tubes in the assembly.

4.C.3.5 All tubes in the test block array shall be from the same heat and shall be installed using identical procedures.

a) The finished thickness of the test block may be less but not greater than the tubesheet it represents. For expanded joints, made with or without welding, the expanded area of the tubes in the test block may be less but not greater than that for the production joint to be qualified.

b) The length of the tube used for testing the tube joint need only be sufficient to suit the test apparatus. The length of the tubes adjacent to the tube joint to be tested shall not be less than the thickness of the test block to be qualified.

4.C.3.6 The procedure used to prepare the tube-to-tubesheet joints in the test specimens shall be the same as used for production.

4.C.3.7 The tube-to-tubesheet joint specimens shall be loaded until mechanical failure of the joint or tube occurs. The essential requirement is that the load be transmitted axially.

4.C.3.8 Any speed of testing may be used provided load readings can be determined accurately.

4.C.3.9 The reading from the testing device shall be such that the applied load required to produce mechanical failure of the tube-to-tubesheet joint can be determined.

4.C.3.10 For determining $f_{r,test}$ for joint types listed in Table 4.C.1, a minimum of three specimens shall constitute a test. The value of $f_{r,test}$ shall be calculated in accordance with paragraph 4.C.4.1 using the lowest value of L_{test}. In no case shall the value of $f_{r,test}$ using a three specimen test exceed the value of $f_{r,test}$ given in Table 4.C.1. If the value of $f_{r,test}$ so determined is less than the value for $f_{r,test}$ given in Table 4.C.1, retesting shall be performed in accordance with paragraph 4.C.3.11, or a new three specimen test shall be performed using a new joint configuration or fabrication procedure. All previous test data shall be rejected. To use a value of $f_{r,test}$ greater than the value given in Table 4.C.1, a nine specimen test shall be performed in accordance with paragraph 4.C.3.11.

4.C.3.11 For joint types not listed in Table 4.C.1, to increase the value of $f_{r,test}$ for joint types listed in Table 4.C.1, or to retest joint types listed in Table 4.C.1, the tests to determine $f_{r,test}$ shall conform to the following.

a) A minimum of nine specimens from a single tube shall be tested. Additional tests of specimens from the same tube are permitted provided all test data are used in the determination of $f_{r,test}$. If a change in the

joint design or its manufacturing procedure is necessary to meet the desired characteristics, complete testing of the modified joint shall be performed.

b) In determining the value of $f_{r,test}$, the mean value of L_{test} shall be determined and the standard deviation, sigma, about the mean shall be calculated. The value of $f_{r,test}$ shall be calculated using the value of L_{test} corresponding to -2 sigma, using the applicable equation in paragraph 4.C.4. In no case shall $f_{r,test}$ exceed 1.0.

4.C.4 Acceptance Standards for Joint Efficiency Factor Determined By Test

4.C.4.1 The value of $f_{r,test}$ determined by testing shall be calculated as follows.

$$f_{r,test} = \frac{L_{test}}{A_t S_T} \qquad \textit{Joint Types a,b,b-1,e} \tag{4.C.7}$$

$$f_{r,test} = \frac{L_{test}}{A_t S_T f_e f_y} \qquad \textit{Joint Types } f,g,h,i,j,k \tag{4.C.8}$$

4.C.4.2 The value of $f_{r,test}$ shall be used for f_r in the equation for $L_{\max}$.

4.C.5 Acceptance Standards for Proposed Operating Temperatures Determined By Test

The proposed operating conditions shall be acceptable if both of the following conditions are satisfied.

$$L_{1,test} \geq A_t f_e f_y S_T \tag{4.C.9}$$

$$L_{2,test} \geq A_t f_e f_y S_T \tag{4.C.10}$$

4.C.6 Nomenclature

A_t tube cross-sectional area.

d_o nominal tube outside diameter.

f_e factor for the length of the expanded portion of the tube. $f_e = \min\left[\left(l/d_o\right), 1.0\right]$ for tube joints made with expanded tubes in tube holes without enhancement and $f_e = 1.0$ for tube joints made with expanded tubes in tube holes with enhancement. An expanded joint is a joint between the tube and tubesheet produced by applying an expanding force inside the portion of the tube to be engaged in the tubesheet. The expanding force shall be set to values necessary to effect sufficient residual interface pressure between the tube and hole for joint strength.

f_r factor to define the efficiency of joint, set equal to the value of $f_{r,test}$ or $f_{r,notest}$. $f_{r,test}$ is equal to the value calculated from results of test in accordance with 4.C.4 or as tabulated in Table 4.C.1, whichever is less, except as permitted in paragraph 4.C.3.11. $f_{r,notest}$ is equal to maximum allowable value without qualification test in accordance with Table 4.C.1.

$f_{r,test}$ factor to define the efficiency of joint established in a test.

$f_{r,notest}$ factor to define the efficiency of joint established without a test.

f_T factor to account for the increase or decrease of tube joint strength due to radial differential thermal expansion at the tube-to-tubesheet joint.

f_y factor for differences in the mechanical properties of tubesheet and tube materials. $f_y = \min\left[\left(S_{y,ts}/S_{y,t}\right),\ 1.0\right]$ for expanded joints. When f_y is less than 0.60, qualification tests in accordance with paragraphs 4.C.3 and 4.C.4 are required.

k tube load factor.
$=1.0$ for loads due to pressure-induced axial forces
$=1.0$ for loads due to thermally-induced or pressure plus thermally-induced axial forces on welded-only joints where the thickness through the weld throat is less than the nominal tube wall thickness t.
$=2.0$ for loads due to thermally-induced or pressure plus thermally-induced axial forces on all other tube-to-tubesheet joints.

l expanded tube length.

$L_{\max}$ maximum allowable axial load in either direction on the tube-to-tubesheet joint.

L_{test} axial load at which failure of the test specimens occur.

$L_{1,test}$ lowest axial load at which failure occurs at operating temperature.

$L_{2,test}$ lowest axial load at which failure of heat soaked specimen tested at room temperature occurs.

P_o interface pressure between the tube and tubesheet that remains after expanding the tube at fabrication. This pressure may be established analytically or experimentally, but shall consider the effect of change in material strength at operating temperature.

P_T interface pressure between the tube and tubesheet due to differential thermal growth. This pressure may be established analytically or experimentally.

S allowable stress from Annex 3.A for the tube at the design temperature. For a welded tube, S is the equivalent allowable stress for a seamless tube.

S_a modified tube allowable stress.

$S_{y,t}$ tube specified minimum yield strength at the design temperature from Annex 3.D.

$S_{y,ts}$ tubesheet specified minimum yield strength at the design temperature from Annex 3.D.

S_T tensile strength for tube material from the material test report

S_y tensile strength for tube material at operating temperature from Annex3.D.

S_{ua} tensile strength for tube material at room temperature from Annex 3.D.

t nominal tube wall thickness.

4.C.7 Tables

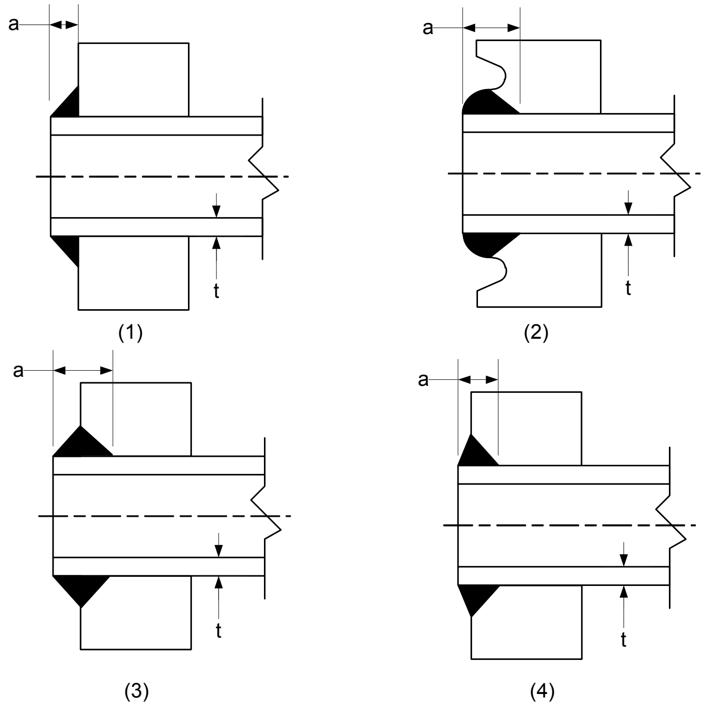
Table 4.C.1 – Efficiencies for Welded and/or Expanded Tube-To-Tubesheet Joints

Joint Type	Description (1)	Notes	$f_{r,test}$ (2)	$f_{r,notest}$
a	Welded only, $a \geq 1.4t$	3	1.00	0.80
b	Welded only, $t \leq a < 1.4t$	3	0.70	0.55
b-1	Welded only, $a < t$	4	0.70	---
e	Welded, $a \geq 1.4t$, and expanded	3	1.00	0.80
f	Welded, $a < 1.4t$, and expanded Enhanced with two or more grooves	3, 5, 6, 7	0.95	0.75
g	Welded, $a < 1.4t$, and expanded Enhanced with single groove	3, 5, 6, 7	0.85	0.65
h	Welded, $a < 1.4t$, and expanded Not enhanced	3, 5, 6	0.70	0.50
i	Expanded, Enhanced with two or more grooves	5, 6, 7	0.90	0.70
j	Expanded Enhanced with single groove	5, 6, 7	0.80	0.65
k	Expanded Not enhanced	5, 6	0.60	0.50

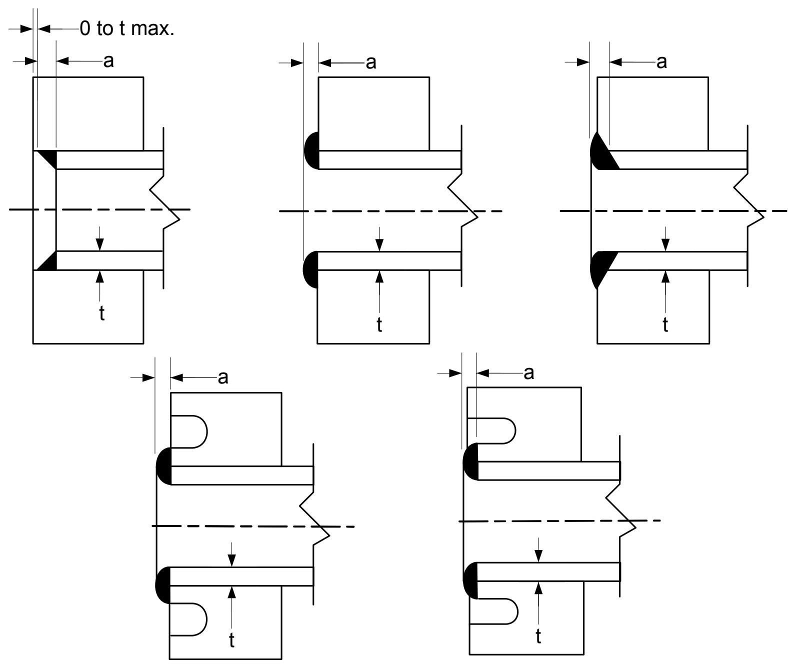
Notes:

1) For joint types involving more than one fastening method, the sequence used in the joint description does not necessarily indicate the order in which the operations are performed.
2) The use of the $f_{r,test}$ factor requires qualification in accordance with paragraphs 4.C.3 and 4.C.4.
3) The value of $f_{r,notest}$ applies only to material combinations as provided for under Section IX. For material combinations not provided for under Section IX, f_r shall be determined by test in accordance with paragraphs 4.C.3 and 4.C.4.
4) For $f_{r,notest}$, refer to paragraph 4.18.10.2.b.
5) If $d_o/(d_o - 2t) < 1.05$ or $d_o/(d_o - 2t) > 1.410$, f_r shall be determined by test in accordance with paragraphs 4.C.3 and 4.C.4.
6) If the nominal pitch (center-to-center distance of adjacent tube holes) is less than $d_o + 2t$, f_r shall be determined by test in accordance with paragraphs 4.C.3 and 4.C.4.
7) The Manufacturer may use other means to enhance the strength of expanded joints, provided however, that the joints are tested in accordance with paragraphs 4.C.3 and 4.C.4.

4.C.8 Figures



Some acceptable weld geometries where a is greater than or equal to 1.4t



Some acceptable weld geometries where a is less than 1.4t

Figure 4.C.1 – Some Acceptable Types of Tube-To-Tubesheet Joints

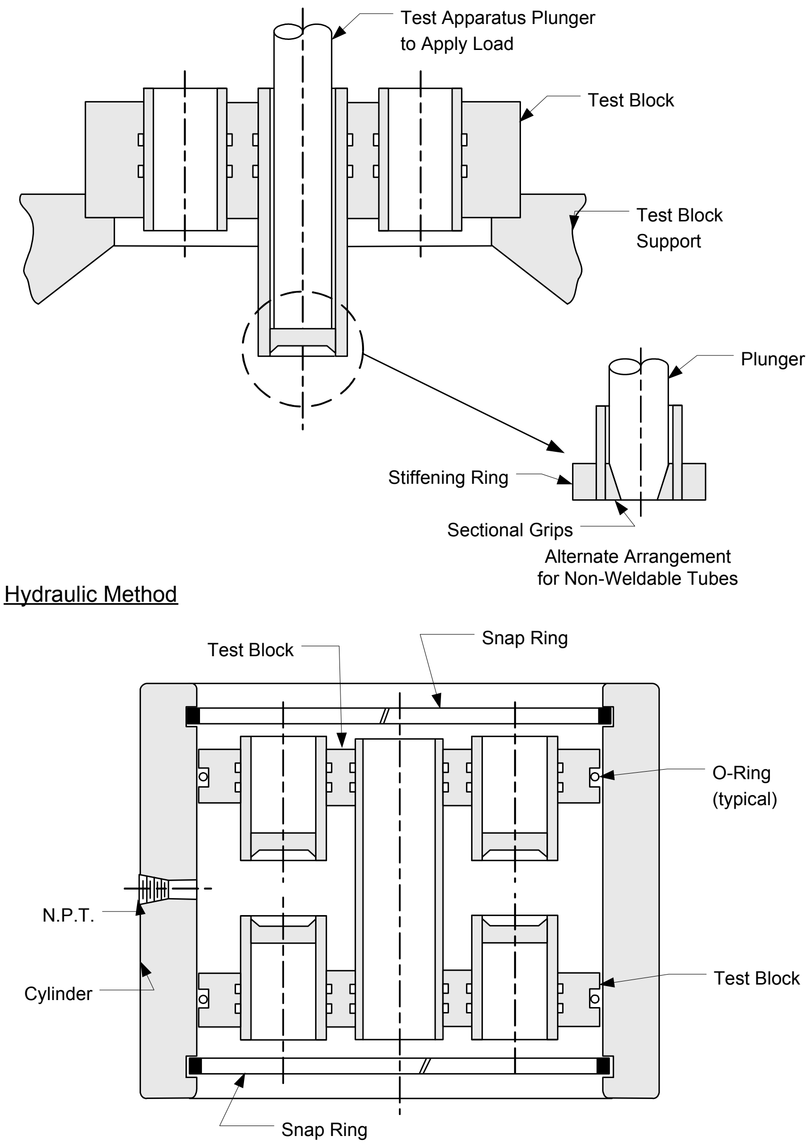


Figure 4.C.2 –Typical Test Fixtures for Expanded or Welded Tube-To-Tubesheet Joints

ANNEX 4.D

GUIDANCE TO ACCOMMODATE LOADINGS PRODUCED BY DEFLAGRATION

(NORMATIVE)

4.D.1 Scope

When an internal vapor-air or dust-air deflagration is defined by the user or his designated agent as a load condition to be considered in the design, this Annex provides guidance for the designer to enhance the ability of a pressure vessel to withstand the forces produced by such conditions.

4.D.2 General

Deflagration is the propagation of a combustion zone at a velocity that is less than the speed of sound in the unreacted medium, whereas detonation is the propagation of a combustion zone at a velocity that is greater than the speed of sound in the unreacted medium. A detonation can produce significant dynamic effects in addition to pressure increases of great magnitude and very short duration, and is outside the scope of this Annex. This Annex only addresses the lower and slower loadings produced by deflagrations that propagate in a gas-phase.

The magnitude of the pressure rise produced inside the vessel by a deflagration is predictable with reasonable certainty. Unvented deflagration pressures can be predicted with more certainty then vented deflagration pressures. Methods are provided in the references listed in 4.D.5 to bound this pressure rise. Other methods may also be used to determine pressure rise.

4.D.3 Design Limitations

The limits of validity for deflagration pressure calculations are described in References [1] and [2].

4.D.4 Design Criteria

4.D.4.1 Safety Margin

As described in NFPA-69, (see Reference [1]), a vessel may be designed to withstand the loads produced by a deflagration:

a) without significant permanent deformation; or

b) without rupture (see Reference [3]).

A decision between these two design criteria should be made by the user or his designated agent based upon the likelihood of the occurrence and the consequences of significant deformation. It is noted that either (a) or (b) above will result in stresses for a deflagration that are larger than the basic Code allowable stress listed in Section II, Part D. Because of this, appropriate design details and nondestructive examination requirements shall be agreed upon between the user and designer.

These two criteria are very similar in principle to the Level C and Level D criteria, respectively, contained in Section III, Subsection NB for use with Class 1 vessels, (see References [4] and [5]). The limited guidance in NFPA 69 requires the application of technical judgments made by knowledgeable designers experienced in the selection and design of appropriate details. The Level C and Level D criteria in Section III provide the detailed methodology for design and analysis. The successful use of either NFPA 69 or Section III criteria for deflagration events requires the selection of materials for construction the will not fail because of brittle fracture during the deflagration pressure excursions.

4.D.4.2 Likelihood of Occurrence

For vapor-air and dust-air combustion, various methods of reducing the likelihood of occurrence are described in Reference [2]. It is good engineering practice to minimize the likelihood of occurrence of these events, regardless of the capability of the vessel to withstand them.

4.D.4.3 Consequences of Occurrence

In deciding between designing to prevent significant permanent deformation (see 4.D.4.1.a) or designing to prevent rupture (see 4.D.4.1.b), the consequences of significant distortion of the pressure boundary should be considered. Either the aforementioned NFPA or Section III design criteria may be used: Each has been used successfully.

4.D.4.4 Strain Concentration

When developing a design to withstand either of the criteria sited above, the designer should avoid creating weak sections in the vessel at which strain can be concentrated. Examples of design details to avoid are partial penetration pressure boundary welds, cone to cylinder junctions without transition knuckles, large openings in heads or cylindrical shells which require special design consideration etc.

4.D.5 References

1. National Fire Protection Association (NFPA) 69, Standard on Explosion Prevention Systems, Chapter 5, Deflagration Pressure Containment, issue effective with the applicable Addenda of the ASME Boiler and Pressure Vessel Code.
2. National Fire Protection Association (NFPA) 68, Guide for the Venting of Deflagrations, issue effective with the applicable Addenda of the ASME Boiler and Pressure Vessel Code.
3. B.F. Langer, PVRC Interpretive Report of Pressure Vessel Research, Section 1- Design Considerations, 1.4 Bursting Strength, Welding Research Council Bulletin 95, April 1964.
4. ASME Boiler and Pressure Vessel Code, Section III, Division 1, NB-3224, Level C Service Limits.
5. ASME Boiler and Pressure Vessel Code, Section III, Division 1, NB-3225 and Appendix F, Level D Service Limits.

PART 5

DESIGN BY ANALYSIS REQUIREMENTS

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5.1 General Requirements

5.1.1 Scope

5.1.1.1 The design requirements for application of the design-by-analysis methodology of this Division are described in Part 5. Detailed design procedures utilizing the results from a stress analysis are provided to evaluate components for plastic collapse, local failure, buckling, and cyclic loading. Supplemental requirements are provided for the analysis of bolts, perforated plates and layered vessels. Procedures are also provided for design using the results from an experimental stress analysis, and for fracture mechanics evaluations.

5.1.1.2 The design-by-analysis requirements are organized based on protection against the failure modes listed below. The component shall be evaluated for each applicable failure mode. If multiple assessment procedures are provided for a failure mode, only one of these procedures must be satisfied to qualify the design of a component.

a) Protection Against Plastic Collapse – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules.

b) Protection Against Local Failure – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules. It is not necessary to evaluate the local strain limit criterion if the component design is in accordance with Part 4 (i.e. component wall thickness and weld detail per paragraph 4.2).

c) Protection Against Collapse From Buckling – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules and the applied loads result in a compressive stress field.

d) Protection Against Failure From Cyclic Loading – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules and the applied loads are cyclic. In addition, these requirements can also be used to qualify a component for cyclic loading where the thickness and size of the component are established using the design-by-rule requirements of Part 4.

5.1.1.3 The design-by-analysis procedures in Part 5 may only be used if the allowable stress from Annex 3.A evaluated at the design temperature is governed by time-independent properties unless otherwise noted in a specific design procedure. If the allowable stress from Annex 3.A evaluated at the design temperature is governed by time-dependent properties and the fatigue screening criteria of paragraph 5.5.2.2 are satisfied, the elastic stress analysis procedures in paragraphs 5.2.2, 5.3.2, 5.6, 5.7.1, 5.7.2, and 5.8 may be used.

5.1.2 Numerical Analysis

5.1.2.1 The design-by-analysis rules in Part 5 are based on the use of results obtained from a detailed stress analysis of a component. Depending on the loading condition, a thermal analysis to determine the temperature distribution and resulting thermal stresses may also be required.

5.1.2.2 Procedures are provided for performing stress analyses to determine protection against plastic collapse, local failure, buckling, and cyclic loading. These procedures provide the necessary details to obtain a consistent result with regards to development of loading conditions, selection of material properties, post-processing of results, and comparison to acceptance criteria to determine the suitability of a component.

5.1.2.3 Recommendations on a stress analysis method, modeling of a component, and validation of analysis results are not provided. While these aspects of the design process are important and shall be considered in the analysis, a detailed treatment of the subject is not provided because of the variability in approaches and design processes. However, an accurate stress analysis including validation of all results shall be provided as part of the design.

5.1.2.4 The following material properties for use in the stress analysis shall be determined using the data and material models in Part 3.

a) Physical properties – Young's Modulus, thermal expansion coefficient, thermal conductivity, thermal diffusivity, density, Poisson's ratio

b) Strength Parameters – Allowable stress, minimum specified yield strength, minimum specified tensile strength

c) Monotonic Stress-Strain Curve – elastic perfectly plastic and elastic-plastic true stress-strain curve with strain hardening

d) Cyclic Stress-Strain Curve – Stabilized true stress-strain amplitude curve

5.1.3 Loading Conditions

5.1.3.1 All applicable applied loads on the component shall be considered when performing a design-by-analysis. Supplemental loads shall be considered in addition to the applied pressure in the form of applicable load cases. If the load case varies with time, a loading histogram shall be developed to show the time variation of each specific load. The load case definition shall be included in the User's Design Specification. An overview of the supplemental loads and loading conditions that shall be considered in a design are shown in Table 5.1.

5.1.3.2 Load case combinations shall be considered in the analysis. Typical load descriptions are provided in Table 5.2. Load case combinations for elastic analysis, limit load analysis, and elastic plastic analysis are shown in Tables 5.3, 5.4, and 5.5, respectively. In evaluating load cases involving the pressure term, P, the effects of the pressure being equal to zero shall be considered. The applicable load case combinations shall be considered in addition to any other combinations defined in the User's Design Specification.

5.1.3.3 If any of the loads vary with time, a loading histogram shall be developed to show the time variation of each specific load. The loading histogram shall include all significant operating temperatures, pressures, supplemental loads, and the corresponding cycles or time periods for all significant events that are applied to the component. The following shall be considered in developing the loading histogram.

a) The number of cycles associated with each event during the operation life, these events shall include start-ups, normal operation, upset conditions, and shutdowns.

b) When creating the histogram, the history to be used in the assessment shall be based on the anticipated sequence of operation. When it is not possible or practical to develop a histogram based on the actual sequence of operation, a histogram may be used that bounds the actual operation. Otherwise, the cyclic evaluation shall account for all possible combinations of loadings.

c) Applicable loadings such as pressure, temperature, supplemental loads such as weight, support displacements, and nozzle reaction loadings.

d) The relationship between the applied loadings during the time history.

5.2 Protection Against Plastic Collapse

5.2.1 Overview

5.2.1.1 Three alternative analysis methods are provided for evaluating protection against plastic collapse. A brief description of these analysis methodologies is provided below.

a) Elastic Stress Analysis Method – Stresses are computed using an elastic analysis, classified into categories, and limited to allowable values that have been conservatively established such that a plastic collapse will not occur.

b) Limit-Load Method – A calculation is performed to determine a lower bound to the limit load of a component. The allowable load on the component is established by applying design factors to the limit load such that the onset of gross plastic deformations (plastic collapse) will not occur.

c) Elastic-Plastic Stress Analysis Method – A collapse load is derived from an elastic-plastic analysis considering both the applied loading and deformation characteristics of the component. The allowable load on the component is established by applying design factors to the plastic collapse load.

5.2.1.2 For components with a complex geometry and/or complex loading, the categorization of stresses requires significant knowledge and judgment. This is especially true for three-dimensional stress fields. Application of the limit load or elastic-plastic analysis methods in paragraphs 5.2.3 and 5.2.4, respectively, is recommended for cases where the categorization process may produce ambiguous results.

5.2.1.3 The use of elastic stress analysis combined with stress classification procedures to demonstrate structural integrity for heavy-wall $\left(R/t \le 4\right)$ pressure containing components, especially around structural discontinuities, may produce non-conservative results and is not recommended. The reason for the non-conservatism is that the nonlinear stress distributions associated with heavy wall sections are not accurately represented by the implicit linear stress distribution utilized in the stress categorization and classification procedure. The misrepresentation of the stress distribution is enhanced if yielding occurs. For example, in cases where calculated peak stresses are above yield over a through thickness dimension which is more than five percent of the wall thickness, linear elastic analysis may give a non-conservative result. In these cases, the elastic-plastic stress analysis procedures in paragraphs 5.2.3 or 5.2.4 shall be used.

5.2.1.4 The structural evaluation procedures based on elastic stress analysis in paragraph 5.2.2 provide an approximation of the protection against plastic collapse. A more accurate estimate of the protection against plastic collapse of a component can be obtained using elastic-plastic stress analysis to develop limit and plastic collapse loads. The limits on the general membrane equivalent stress, local membrane equivalent stress and primary membrane plus primary bending equivalent stress in paragraph 5.2.2 have been placed at a level which conservatively assures the prevention of collapse as determined by the principles of limit analysis. These limits need not be satisfied if the requirements of paragraph 5.2.3 or paragraph 5.2.4 are satisfied.

5.2.2 Elastic Stress Analysis Method

5.2.2.1 Overview

To evaluate protection against plastic collapse, the results from an elastic stress analysis of the component subject to defined loading conditions are categorized and compared to an associated limiting value. The basis of the categorization procedure is described below.

a) A quantity known as the equivalent stress is computed at locations in the component and compared to an allowable value of equivalent stress to determine if the component is suitable for the intended design conditions. The equivalent stress at a point in a component is a measure of stress, calculated from stress components utilizing a yield criterion, which is used for comparison with the mechanical strength properties of the material obtained in tests under uniaxial load.

b) The maximum distortion energy yield criterion shall be used to establish the equivalent stress. In this case, the equivalent stress is equal to the von Mises equivalent stress given by Equation.(5.1)

$$S_e = \sigma_e = \frac{1}{\sqrt{2}}\left[\left(\sigma_1-\sigma_2\right)^2+\left(\sigma_2-\sigma_3\right)^2+\left(\sigma_3-\sigma_1\right)^2\right]^{0.5} \quad (5.1)$$

5.2.2.2 Stress Categorization

The three basic equivalent stress categories and associated limits that are to be satisfied for plastic collapse are defined below. The terms general primary membrane stress, local primary membrane stress, primary

bending stress, secondary stress, and peak stress used for elastic analysis are defined in the following paragraphs. The design loads to be evaluated and the allowable stress limits are provided in Table 5.3. Stress limits for the pressure test condition are covered in paragraph 4.1.6.2.

a) General Primary Membrane Equivalent Stress (P_m)

1) The general primary membrane equivalent stress (see Figure 5.1) is the equivalent stress, derived from the average value across the thickness of a section, of the general primary stresses produced by the design internal pressure and other specified mechanical loads but excluding all secondary and peak stresses.

2) Examples of this stress category for typical pressure vessel components are shown in Table 5.6.

b) Local Primary Membrane Equivalent Stress (P_L)

1) The local primary membrane equivalent stress (see Figure 5.1) is the equivalent stress, derived from the average value across the thickness of a section, of the local primary stresses produced by the design pressure and specified mechanical loads but excluding all secondary and peak stresses. A region of stress in a component is considered as local if the distance over which the equivalent stress exceeds $1.1S$ does not extend in the meridional direction more than $\sqrt{Rt}$.

2) Regions of local primary membrane stress that exceed $1.1S$ shall be separated in the meridional direction by a distance greater than or equal to $1.25\sqrt{(R_1+R_2)(t_1+t_2)}$. Discrete regions of local primary membrane stress, such as those resulting from concentrated loads on support brackets, where the membrane stress exceeds $1.1S$, shall be spaced so that there is not an overlapping area in which the membrane stress exceeds $1.1S$.

3) Examples of this stress category for typical pressure vessel components are shown in Table 5.6.

c) Primary Membrane (General or Local) Plus Primary Bending Equivalent Stress (P_L+P_b)

1) The Primary Membrane (General or Local) Plus Primary Bending Equivalent Stress (see Figure 5.1) is the equivalent stress, derived from the highest value across the thickness of a section, of the linearized general or local primary membrane stresses plus primary bending stresses produced by design pressure and other specified mechanical loads but excluding all secondary and peak stresses.

2) Examples of this stress category for typical pressure vessel components are shown in Table 5.6.

5.2.2.3 Linearization of Stress Results for Stress Classification

Results from an elastic stress analysis can be used to compute the equivalent linearized membrane and bending stresses for comparison to the limits in paragraph 5.2.2.4 using the methods described in Annex 5.A.

5.2.2.4 Assessment Procedure

To determine the acceptability of a component, the computed equivalent stresses given in paragraph 5.2.2.2 for a component subject to loads shall not exceed the specified allowable values. A schematic illustrating the categorization of equivalent stresses and their corresponding allowable values is shown in Figure 5.1. The following procedure is used to compute and categorize the equivalent stress at a point in a component (see paragraph 5.2.2.3), and to determine the acceptability of the resulting stress state.

a) STEP 1 – Determine the types of loads acting on the component. In general, separate load cases are analyzed to evaluate "load-controlled" loads such as pressure and externally applied reactions due to weight effects and "strain-controlled" loads resulting from thermal gradients and imposed displacements. The loads to be considered in the design shall include, but not be limited to, those given in Table 5.1.

The load combinations that shall be considered for each loading condition shall include, but not be limited to those given in Table 5.3.

b) STEP 2 – At the point on the vessel that is being investigated, calculate the stress tensor (six unique components of stress) for each type of load. Assign each of the computed stress tensors to one or to a group of the categories defined below. Assistance in assigning each stress tensor to an appropriate category for a component can be obtained by using Figure 5.1 and Table 5.6. Note that the equivalent stresses Q and F do not need to be determined to evaluate protection against plastic collapse. However, these components are needed for fatigue and ratcheting evaluations that are based on elastic stress analysis (see paragraphs 5.5.3 and 5.5.6, respectively).

   1) General primary membrane equivalent stress – P_m

   2) Local primary membrane equivalent stress – P_L

   3) Primary bending equivalent stress – P_b

   4) Secondary equivalent stress – Q

   5) Additional equivalent stress produced by a stress concentration or a thermal stress over and above the nominal $(P+Q)$ stress level – F

c) STEP 3 – Sum the stress tensors (stresses are added on a component basis) assigned to each equivalent stress category. The final result is a stress tensor representing the effects of all the loads assigned to each equivalent stress category. Note that in applying STEPs in this paragraph, a detailed stress analysis performed using a numerical method such as finite element analysis typically provides a combination of $P_L + P_b$ and $P_L + P_b + Q + F$ directly.

   1) If a load case is analyzed that includes only "load-controlled" loads (e.g. pressure and weight effects), the computed equivalent stresses shall be used to directly represent the P_m, $P_L + P_b$, or $P_L + P_b + Q$. For example, for a vessel subject to internal pressure with an elliptical head; P_m equivalent stresses occur away from the head to shell junction, and P_L and $P_L + P_b + Q$ equivalent stresses occur at the junction.

   2) If a load case is analyzed that includes only "strain-controlled" loads (e.g. thermal gradients), the computed equivalent stresses represent Q alone; the combination $P_L + P_b + Q$ shall be derived from load cases developed from both "load-controlled" and "strain-controlled" loads.

   3) If the stress in category F is produced by a stress concentration or thermal stress, the quantity F is the additional stress produced by the stress concentration in excess of the nominal membrane plus bending stress. For example, if a plate has a nominal primary membrane equivalent stress of S_e, and has a fatigue strength reduction characterized by a factor K_f, then: $P_m = S_e$, $P_b = 0$, $Q = 0$, and $F = P_m\left(K_f - 1\right)$. The total equivalent stress is $P_m + F$.

d) STEP 4 – Determine the principal stresses of the sum of the stress tensors assigned to the equivalent stress categories, and compute the equivalent stress using Equation (5.1).

e) STEP 5 – To evaluate protection against plastic collapse, compare the computed equivalent stress to their corresponding allowable values (see paragraph 5.2.2.2).

$$P_m \le S \tag{5.2}$$

$$P_L \le 1.5S \tag{5.3}$$

$$\left(P_L + P_b\right) \le 1.5S \tag{5.4}$$

5.2.3 Limit-Load Analysis Method

5.2.3.1 Overview

a) Limit-load analysis addresses the failure modes of ductile rupture and the onset of gross plastic deformation (plastic collapse) of a structure. As defined in the following paragraphs, it provides one option to protect a vessel or component from plastic collapse. It is to be applied to single or multiple static loading, applied in any specified order. Limit-load analysis provides an alternative to elastic analysis and stress linearization and the satisfaction of primary stress limits in paragraphs 5.2.2.2.

b) Displacements and strains indicated by a limit analysis solution have no physical meaning. If the User's Design Specification requires a limit on such variables, the procedures in paragraph 5.2.4 shall be used to satisfy these requirements.

c) Protection against plastic collapse using limit load analysis is based on the theory of limit analysis that defines a lower bound to the limit load of a structure as the solution of a numerical model with the following properties:

   1) The material model is elastic-perfectly plastic with a specified yield strength.

   2) The strain-displacement relations are those of small displacement theory.

   3) Equilibrium is satisfied in the undeformed configuration.

5.2.3.2 Limitations

The following limitations apply equally to limit-load analysis and to primary stress limits of paragraph 5.2.2.

a) The effect of strain-controlled loads resulting from prescribed non-zero displacements and temperature fields is not considered.

b) Components that experience reduction in stiffness with deformation, e.g. a pipe elbow under in-plane bending, shall be evaluated using paragraph 5.2.4.

5.2.3.3 Numerical Analysis

The limit load is obtained using a numerical analysis technique (e.g. finite element method) by incorporating an elastic-perfectly-plastic material model and small displacement theory to obtain a solution. The limit load is the load that causes overall structural instability. This point is indicated by the inability to achieve an equilibrium solution for a small increase in load (i.e. the solution will not converge).

5.2.3.4 Acceptance Criteria

The acceptability of a component using a limit-load analysis is determined by satisfying the following two criteria.

a) Global Criteria – A global plastic collapse load is established by performing a limit-load analysis of the component subject to the specified loading conditions. The plastic collapse load is taken as the load which causes overall structural instability. The concept of Load and Resistance Factor Design (LRFD) is used as an alternative to the rigorous computation of a plastic collapse load to design a component. In this procedure, factored loads that include a design factor to account for uncertainty, and the resistance of the component to these factored loads is determined using a limit load analysis (see Table 5.4).

b) Service Criteria – Service criteria as provided by the Owner/User that limit the potential for unsatisfactory performance shall be satisfied at every location in the component when subject to the design loads. The service criteria shall satisfy the requirements of paragraph 5.2.4.3.b using the procedures in paragraph 5.2.4.

5.2.3.5 Assessment Procedure

The following assessment procedure is used to determine the acceptability of a component using a limit-load analysis.

a) STEP 1 – Develop a numerical model of the component including all relevant geometry characteristics. The model used for the analysis shall be selected to accurately represent the component geometry, boundary conditions, and applied loads. The model need not be accurate for small details, such as small holes, fillets, corner radii, and other stress raisers, but should otherwise correspond to commonly accepted practice.

b) STEP 2 – Define all relevant loads and applicable load cases. The loads to be considered in the analysis shall include, but not be limited to, those given in Table 5.1.

c) STEP 3 – An elastic-perfectly plastic material model with small displacement theory shall be used in the analysis. The von Mises yield function and associated flow rule should be utilized. The yield strength defining the plastic limit shall equal $1.5S$.

d) STEP 4 – Determine the load case combinations to be used in the analysis using the information from STEP 2 in conjunction with Table 5.4. Each of the indicated load cases shall be evaluated. The effects of one or more loads not acting shall be investigated. Additional load cases for special conditions not included in Table 5.4 shall be considered, as applicable.

e) STEP 5 – Perform a limit-load analysis for each of the load case combinations defined in STEP 4. If convergence is achieved, the component is stable under the applied loads for this load case. Otherwise, the component configuration (i.e. thickness) shall be modified or applied loads reduced and the analysis repeated. Note that if the applied loading results in a compressive stress field within the component, buckling may occur, and the effects of imperfections, especially for shell structures, should be considered in the analysis (see paragraph 5.4).

5.2.4 Elastic-Plastic Stress Analysis Method

5.2.4.1 Overview

a) Protection against plastic collapse is evaluated by determining the plastic collapse load of the component using an elastic-plastic stress analysis. The allowable load on the component is established by applying a design factor to the calculated plastic collapse load.

b) Elastic-plastic stress analysis provides a more accurate assessment of the protection against plastic collapse of a component relative to the criteria in paragraphs 5.2.2 and 5.2.3 because the actual structural behavior is more closely approximated. The redistribution of stress that occurs as a result of inelastic deformation (plasticity) and deformation characteristics of the component are considered directly in the analysis.

5.2.4.2 Numerical Analysis

The plastic collapse load can be obtained using a numerical analysis technique (e.g. finite element method) by incorporating an elastic-plastic material model to obtain a solution. The effects of non-linear geometry shall be considered in this analysis. The plastic collapse load is the load that causes overall structural instability. This point is indicated by the inability to achieve an equilibrium solution for a small increase in load (i.e. the solution will not converge).

5.2.4.3 Acceptance Criteria

The acceptability of a component using an elastic-plastic analysis is determined by satisfying the following two criteria.

a) Global Criteria – A global plastic collapse load is established by performing an elastic-plastic analysis of the component subject to the specified loading conditions. The plastic collapse load is taken as the load which causes overall structural instability. The concept of Load and Resistance Factor Design (LRFD) is

used as an alternate to the rigorous computation of a plastic collapse load to design a component. In this procedure, factored loads that include a design factor to account for uncertainty, and the resistance of the component to these factored loads are determined using an elastic-plastic analysis (see Table 5.5).

b) Service Criteria – Service criteria that limit the potential for unsatisfactory performance shall be satisfied at every location in the component when subject to the design loads (see Table 5.5). Examples of service criteria are limits on the rotation of a mating flange pair to avoid possible flange leakage concerns and limits on tower deflection that may cause operational concerns. In addition, the effect of deformation of the component on service performance shall be evaluated at the design load combinations. This is especially important for components that experience an increase in resistance (geometrically stiffen) with deformation under applied loads such as elliptical or torispherical heads subject to internal pressure loading. The plastic collapse criteria may be satisfied but the component may have excessive deformation at the derived design conditions. In this case, the design loads may have to be reduced based on a deformation criterion. Examples of some of the considerations in this evaluation are the effect of deformation on:

1) piping connections or,

2) misalignment of trays, platforms and other internal or external appurtenances, and

3) interference with adjacent structures and equipment.

If applicable, the service criteria shall be specified in the User's Design Specification.

5.2.4.4 Assessment Procedure

The following assessment procedure is used to determine the acceptability of a component using an elastic-plastic stress analysis.

a) STEP 1 – Develop a numerical model of the component including all relevant geometry characteristics. The model used for the analysis shall be selected to accurately represent the component geometry, boundary conditions, and applied loads. In addition, refinement of the model around areas of stress and strain concentrations shall be provided. The analysis of one or more numerical models may be required to ensure that an accurate description of the stress and strains in the component is achieved.

b) STEP 2 – Define all relevant loads and applicable load cases. The loads to be considered in the design shall include, but not be limited to, those given in Table 5.1.

c) STEP 3 – An elastic-plastic material model shall be used in the analysis. The von Mises yield function and associated flow rule should be utilized if plasticity is anticipated. A material model that includes hardening or softening, or an elastic-perfectly plastic model may be utilized. A true stress-strain curve model that includes temperature dependent hardening behavior is provided in Appendix 3.D. When using this material model, the hardening behavior shall be included up to the true ultimate stress and perfect plasticity behavior (i.e. the slope of the stress-strain curves is zero) beyond this limit. The effects of non-linear geometry shall be considered in the analysis.

d) STEP 4 – Determine the load case combinations to be used in the analysis using the information from STEP 2 in conjunction with Table 5.5. Each of the indicated load cases shall be evaluated. The effects of one or more loads not acting shall be investigated. Additional load cases for special conditions not included in Table 5.5 shall be considered, as applicable.

e) STEP 5 – Perform an elastic-plastic analysis for each of the load cases defined in STEP 4. If convergence is achieved, the component is stable under the applied loads for this load case. Otherwise, the component configuration (i.e. thickness) shall be modified or applied loads reduced and the analysis repeated. Note that if the applied loading results in a compressive stress field within the component, buckling may occur, and an evaluation in accordance with paragraph 5.4 may be required.

5.3 Protection Against Local Failure

5.3.1 Overview

5.3.1.1 In addition to demonstrating protection against plastic collapse as defined in paragraph 5.2, the applicable local failure criteria below shall be satisfied for a component. The local strain limit criterion does not need to be checked if the component design is in accordance with the standard details of Part 4.

5.3.1.2 Two analysis methodologies are provided for evaluating protection against local failure to limit the potential for fracture under applied design loads.

a) The analysis procedures in paragraph 5.3.2 provide an approximation of the protection against local failure based on the results of an elastic analysis.

b) A more accurate estimate of the protection against local failure of a component can be obtained using the elastic-plastic stress analysis procedures in paragraph 5.3.3.

5.3.2 Elastic Analysis

In addition to demonstrating protection against plastic collapse, the following elastic analysis criterion shall be satisfied for each point in the component. The sum of the local primary membrane plus bending principal stresses shall be used for checking this criterion.

$$\left(\sigma_1+\sigma_2+\sigma_3\right)\le 4S \tag{5.5}$$

5.3.3 Elastic-Plastic Analysis

5.3.3.1 The following procedure shall be used to evaluate protection against local failure for a sequence of applied loads.

a) STEP 1 – Perform an elastic-plastic stress analysis based on the load case combinations for the local criteria given in Table 5.5. The effects of non-linear geometry shall be considered in the analysis.

b) STEP 2 – For a location in the component subject to evaluation, determine the principal stresses, σ_1, σ_2, σ_3, the equivalent stress, σ_e, using Equation (5.1) and the total equivalent plastic strain, ε_{peq}.

c) STEP 3 – Determine the limiting triaxial strain, ε_L, using Equation (5.6) where ε_{Lu}, m_2, and α_{sl} are determined from Table 5.7.

$$\varepsilon_L=\varepsilon_{Lu}\cdot\exp\left[-\left(\frac{\alpha_{sl}}{1+m_2}\right)\left(\left\{\frac{\left(\sigma_1+\sigma_2+\sigma_3\right)}{3\sigma_e}\right\}-\frac{1}{3}\right)\right] \tag{5.6}$$

d) STEP 4 – Determine the forming strain ε_{cf} based on the material and fabrication method in accordance with Part 6. If heat treatment is performed in accordance with Part 6, the forming strain may be assumed to be zero.

e) STEP 5– Determine if the strain limit is satisfied. The location in the component is acceptable for the specified load case if Equation (5.7) is satisfied.

$$\varepsilon_{peq}+\varepsilon_{cf}\le\varepsilon_L \tag{5.7}$$

5.3.3.2 If a specific loading sequence is to be evaluated in accordance with the User's Design Specification, a strain limit damage calculation procedure may be required. This procedure may also be used in lieu of the procedure in paragraph 5.3.3.1. In this procedure, the loading path is divided into k load increments and the principal stresses, $\sigma_{1,k}$, $\sigma_{2,k}$, $\sigma_{3,k}$, equivalent stress, $\Delta\sigma_{e,k}$, and change in the equivalent plastic strain from the previous load increment, $\Delta\varepsilon_{peq,k}$, are calculated for each load increment. The strain limit for the k^{th} load increment, $\varepsilon_{L,k}$, is calculated using Equation (5.8) where ε_{Lu}, m_2, and α_{sl} are determined from Table 5.7. The strain limit damage for each load increment is calculated using Equation (5.9) and the strain limit damage from forming, $D_{\varepsilon form}$, is calculated using Equation (5.10). If heat treatment is performed in accordance with Part 6, the strain limit damage from forming is assumed to be zero. The accumulated strain limit damage is calculated using Equation (5.11). The location in the component is acceptable for the specified loading sequence if this equation is satisfied.

$$\varepsilon_{L,k} = \varepsilon_{Lu} \cdot \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2}\right)\left(\left\{\frac{\left(\sigma_{1,k}+\sigma_{2,k}+\sigma_{3,k}\right)}{3\sigma_{e,k}}\right\}-\frac{1}{3}\right)\right] \tag{5.8}$$

$$D_{\varepsilon,k} = \frac{\Delta\varepsilon_{peq,k}}{\varepsilon_{L,k}} \tag{5.9}$$

$$D_{\varepsilon form} = \frac{\varepsilon_{cf}}{\varepsilon_{Lu} \cdot \exp\left[-0.67\left(\frac{\alpha_{sl}}{1+m_2}\right)\right]} \tag{5.10}$$

$$D_{\varepsilon} = D_{\varepsilon form} + \sum_{k=1}^{M} D_{\varepsilon,k} \le 1.0 \tag{5.11}$$

5.4 Protection Against Collapse From Buckling

5.4.1 Design Factors

5.4.1.1 In addition to evaluating protection against plastic collapse as defined in paragraph 5.2, a design factor for protection against collapse from buckling shall be satisfied to avoid buckling of components with a compressive stress field under applied design loads.

5.4.1.2 The design factor to be used in a structural stability assessment is based on the type of buckling analysis performed. The following design factors shall be the minimum values for use with shell components when the buckling loads are determined using a numerical solution (i.e. bifurcation buckling analysis or elastic-plastic collapse analysis).

a) Type 1 – If a bifurcation buckling analysis is performed using an elastic stress analysis without geometric nonlinearities in the solution to determine the pre-stress in the component, a minimum design factor of $\Phi_B = 2/\beta_{cr}$ shall be used (see paragraph 5.4.1.3). In this analysis, the pre-stress in the component is established based on the loading combinations in Table 5.3.

b) Type 2 – If a bifurcation buckling analysis is performed using an elastic-plastic stress analysis with the effects of non-linear geometry in the solution to determine the pre-stress in the component, a minimum

design factor of $\Phi_B = 1.667/\beta_{cr}$ shall be used (see paragraph 5.4.1.3). In this analysis, the pre-stress in the component is established based on the loading combinations in Table 5.3.

c) Type 3 – If a collapse analysis is performed in accordance with paragraph 5.2.4, and imperfections are explicitly considered in the analysis model geometry, the design factor is accounted for in the factored load combinations in Table 5.5. It should be noted that a collapse analysis can be performed using elastic or plastic material behavior. If the structure remains elastic when subject to the applied loads, the elastic-plastic material model will provide the required elastic behavior, and the collapse load will be computed based on this behavior.

5.4.1.3 The capacity reduction factors, β_{cr}, shown below shall be used unless alternative factors can be developed from published information.

a) For unstiffened or ring stiffened cylinders and cones under axial compression

$$\beta_{cr} = 0.207 \qquad for \quad \frac{D_o}{t} \geq 1247 \tag{5.12}$$

$$\beta_{cr} = \frac{338}{389 + \frac{D_o}{t}} \qquad for \quad \frac{D_o}{t} < 1247 \tag{5.13}$$

b) For unstiffened and ring stiffened cylinders and cones under external pressure

$$\beta_{cr} = 0.80 \tag{5.14}$$

c) For spherical shells and spherical, torispherical, elliptical heads under external pressure

$$\beta_{cr} = 0.124 \tag{5.15}$$

5.4.2 Numerical Analysis

If a numerical analysis is performed to determine the buckling load for a component, all possible buckling mode shapes shall be considered in determining the minimum buckling load for the component. Care should be taken to ensure that simplification of the model does not result in exclusion of a critical buckling mode shape. For example, when determining the minimum buckling load for a ring-stiffened cylindrical shell, both axisymmetric and non-axisymmetric buckling modes shall be considered in determination of the minimum buckling load.

5.5 Protection Against Failure From Cyclic Loading

5.5.1 Overview

5.5.1.1 A fatigue evaluation shall be performed if the component is subject to cyclic operation. The evaluation for fatigue is made on the basis of the number of applied cycles of a stress or strain range at a point in the component. The allowable number of cycles should be adequate for the specified number of cycles as given in the User's Design Specification.

5.5.1.2 Screening criteria are provided in paragraph 5.5.2 that can be used to determine if fatigue analysis is required as part of a design. If the component does not satisfy the screening criteria, a fatigue evaluation shall be performed using the techniques in paragraphs 5.5.3, 5.5.4 or 5.5.5.

5.5.1.3 Fatigue curves are typically presented in two forms: fatigue curves that are based on smooth bar test specimens and fatigue curves that are based on test specimens that include weld details of quality consistent with the fabrication and inspection requirements of this Division.

a) Smooth bar fatigue curves may be used for components with or without welds. The welded joint curves shall only be used for welded joints.

b) The smooth bar fatigue curves are applicable up to the maximum number of cycles given on the curves. The welded joint fatigue curves do not exhibit an endurance limit and are acceptable for all cycles.

c) If welded joint fatigue curves are used in the evaluation, and if thermal transients result in a through-thickness stress difference at any time that is greater than the steady state difference, the number of design cycles shall be determined as the smaller of the number of cycles for the base metal established using either paragraph 5.5.3 or 5.5.4, and for the weld established in accordance with paragraph 5.5.5.

5.5.1.4 Stresses and strains produced by any load or thermal condition that does not vary during the cycle need not be considered in a fatigue analysis if the fatigue curves utilized in the evaluation are adjusted for mean stresses and strains. The design fatigue curves referenced in paragraph 5.5.3 and 5.5.4 are based on smooth bar test specimens and are adjusted for the maximum possible effect of mean stress and strain; therefore, an adjustment for mean stress effects is not required. The fatigue curves referenced in paragraph 5.5.5 are based on welded test specimens and include explicit adjustments for thickness and mean stress effects.

5.5.1.5 Under certain combinations of steady state and cyclic loadings there is a possibility of ratcheting. A rigorous evaluation of ratcheting normally requires an elastic-plastic analysis of the component; however, under a limited number of loading conditions, an approximate analysis can be utilized based on the results of an elastic stress analysis, see paragraph 5.5.6.

5.5.1.6 Protection against ratcheting shall be considered for all operating loads listed in the User's Design Specification and shall be performed even if the fatigue screening criteria are satisfied (see paragraph 5.5.2). Protection against ratcheting is satisfied if one of the following three conditions is met:

a) The loading results in only primary stresses without any cyclic secondary stresses.

b) Elastic Stress Analysis Criteria – Protection against ratcheting is demonstrated by satisfying the rules of paragraph 5.5.6.

c) Elastic-Plastic Stress Analysis Criteria – Protection against ratcheting is demonstrated by satisfying the rules of paragraph 5.5.7.

5.5.2 Screening Criteria for Fatigue Analysis

5.5.2.1 Overview

a) The provisions of this paragraph can be used to determine if a fatigue analysis is required as part of the vessel design. The screening options to determine the need for fatigue analysis are described below. If any one of the screening options is satisfied, then a fatigue analysis is not required as part of the vessel design.

    1) Provisions of paragraph 5.5.2.2, Experience with comparable equipment operating under similar conditions.

    2) Provisions of paragraph 5.5.2.3, Method A based on the materials of construction (limited applicability), construction details, loading histogram, and smooth bar fatigue curve data.

    3) Provisions of paragraph 5.5.2.4, Method B based on the materials of construction (unlimited applicability), construction details, loading histogram, and smooth bar fatigue curve data.

b) The fatigue exemption in accordance with this paragraph is performed on a component or part basis. One component (integral) may be exempt, while another component (non-integral) is not exempt. If any one component is not exempt, then a fatigue evaluation shall be performed for that component.

c) If the specified number of cycles is greater than $(10)^6$, then the screening criteria are not applicable and a fatigue analysis is required.

5.5.2.2 Fatigue Analysis Screening Based On Experience with Comparable Equipment

If successful experience over a sufficient time frame is obtained with comparable equipment subject to a similar loading histogram and addressed in the User's Design Specification (see 2.2.2.1.f), then a fatigue analysis is not required as part of the vessel design. When evaluating experience with comparable equipment operating under similar conditions as related to the design and service contemplated, the possible harmful effects of the following design features shall be evaluated.

a) The use of non-integral construction, such as the use of pad type reinforcements or of fillet welded attachments, as opposed to integral construction

b) The use of pipe threaded connections, particularly for diameters in excess of 70 mm (2.75 in.)

c) The use of stud bolted attachments

d) The use of partial penetration welds

e) Major thickness changes between adjacent members

f) Attachments and nozzles in the knuckle region of formed heads

5.5.2.3 Fatigue Analysis Screening, Method A

The following procedure can only be used for materials with a specified minimum tensile strength that is less than or equal to 552 MPa (80,000 psi).

a) STEP 1 – Determine a load history based on the information in the User's Design Specification. The load history should include all cyclic operating loads and events that are applied to the component.

b) STEP 2 – Based on the load history in STEP 1, determine the expected (design) number of full-range pressure cycles including startup and shutdown, and designate this value as $N_{\Delta FP}$.

c) STEP 3 – Based on the load history in STEP 1, determine the expected number of operating pressure cycles in which the range of pressure variation exceeds 20% of the design pressure for integral construction or 15% of the design pressure for non-integral construction, and designate this value as $N_{\Delta PO}$. Pressure cycles in which the pressure variation does not exceed these percentages of the design pressure and pressure cycles caused by fluctuations in atmospheric conditions do not need to be considered in this evaluation.

d) STEP 4 – Based on the load history in STEP 1, determine the effective number of changes in metal temperature difference between any two adjacent points, ΔT_E, as defined below, and designate this value as $N_{\Delta TE}$. The effective number of such changes is determined by multiplying the number of changes in metal temperature difference of a certain magnitude by the factor given in Table 5.8, and by adding the resulting numbers. In calculating the temperature difference between adjacent points, conductive heat transfer shall be considered only through welded or integral cross sections with no allowance for conductive heat transfer across un-welded contact surfaces (i.e. vessel shell and reinforcing pad).

    1) For surface temperature differences, points are considered to be adjacent if they are within the distance L computed as follows: for shells and dished heads in the meridional or circumferential directions,

$$L = 2.5\sqrt{Rt} \tag{5.16}$$

and for flat plates,

$$L = 3.5a \tag{5.17}$$

2) For through-the-thickness temperature differences, adjacent points are defined as any two points on a line normal to any surface on the component.

e) STEP 5 – Based on the load history in STEP 1, determine the number of temperature cycles for components involving welds between materials having different coefficients of thermal expansion that causes the value of $(\alpha_1 - \alpha_2)\Delta T$ to exceed 0.00034, and designate this value as $N_{\Delta T\alpha}$.

f) STEP 6 – If the expected number of operating cycles from STEPs 2, 3, 4 and 5 satisfy the criterion in Table 5.9, then a fatigue analysis is not required as part of the vessel design. If this criterion is not satisfied, then a fatigue analysis is required as part of the vessel design. Examples of non-integral attachments are: screwed-on caps, screwed-in plugs, shear ring closures, fillet welded attachments, and breech lock closures.

5.5.2.4 Fatigue Analysis Screening, Method B

The following procedure can be used for all materials.

a) STEP 1 – Determine a load history based on the information in the User's Design Specification. The load histogram should include all significant cyclic operating loads and events that the component will be subjected. Note, in Equation (5.18), the number of cycles from the applicable design fatigue curve (see Annex 3.F) evaluated at a stress amplitude of S_e is defined as $N(S_e)$. Also in Equations (5.19) through (5.23), the stress amplitude from the applicable design fatigue curve (see Annex 3.F) evaluated at N cycles is defined as $S_a(N)$.

b) STEP 2 – Determine the fatigue screening criteria factors, C_1 and C_2, based on the type of construction in accordance with Table 5.10, see paragraph 4.2.5.6.j.

c) STEP 3 – Based on the load histogram in STEP 1, determine the design number of full-range pressure cycles including startup and shutdown, $N_{\Delta FP}$. If the following equation is satisfied, proceed to STEP 4; otherwise, a detailed fatigue analysis of the vessel is required.

$$N_{\Delta FP} \le N\left(C_1 S\right) \tag{5.18}$$

d) STEP 4 – Based on the load histogram in STEP 1, determine the maximum range of pressure fluctuation during normal operation, excluding startups and shutdowns, ΔP_N, and the corresponding number of significant cycles, $N_{\Delta P}$. Significant pressure fluctuation cycles are defined as cycles where the pressure range exceeds $S_{as}/3S$ times the design pressure. If the following equation is satisfied, proceed to STEP 5; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta P_N \le \frac{P}{C_1}\left(\frac{S_a\left(N_{\Delta P}\right)}{S}\right) \tag{5.19}$$

e) STEP 5 – Based on the load histogram in STEP 1, determine the maximum temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation, ΔT_N, and the corresponding number of cycles, $N_{\Delta TN}$. If the following equation is satisfied, proceed to STEP 6; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta T_N \leq \left(\frac{S_a\left(N_{\Delta TN}\right)}{C_2 E_{ym}\alpha} \right) \tag{5.20}$$

f) STEP 6 – Based on the load histogram in STEP 1, determine the maximum range of temperature difference fluctuation, ΔT_R, between any two adjacent points (see paragraph 5.5.2.3.d) of the vessel during normal operation, excluding startups and shutdowns, and the corresponding number of significant cycles, $N_{\Delta TR}$. Significant temperature difference fluctuation cycles for this STEP are defined as cycles where the temperature range exceeds $S_{as}/2E_{ym}\alpha$. If the following equation is satisfied, proceed to STEP 7; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta T_R \leq \left(\frac{S_a\left(N_{\Delta TR}\right)}{C_2 E_{ym}\alpha} \right) \tag{5.21}$$

g) STEP 7 – Based on the load histogram in STEP 1, determine the range of temperature difference fluctuation between any two adjacent points (see paragraph 5.5.2.3.d) for components fabricated from different materials of construction during normal operation, ΔT_M, and the corresponding number of significant cycles, $N_{\Delta TM}$. Significant temperature difference fluctuation cycles for this STEP are defined as cycles where the temperature range exceeds $S_{as}\Big/\left[2\left(E_{y1}\alpha_1 - E_{y2}\alpha_2\right)\right]$. If the following equation is satisfied, proceed to STEP 8; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta T_M \leq \left(\frac{S_a\left(N_{\Delta TM}\right)}{C_2\left(E_{y1}\alpha_1 - E_{y2}\alpha_2\right)} \right) \tag{5.22}$$

h) STEP 8 – Based on the load histogram in STEP 1, determine the equivalent stress range computed from the specified full range of mechanical loads, excluding pressure but including piping reactions, ΔS_{ML}, and the corresponding number of significant cycles, $N_{\Delta S}$. Significant mechanical load range cycles for this STEP are defined as cycles where the stress range exceeds S_{as}. If the total specified number of significant load fluctuations exceeds the maximum number of cycles defined on the applicable fatigue curve, the S_{as} value corresponding to the maximum number of cycles defined on the fatigue curve shall be used. If the following equation is satisfied a fatigue analysis is not required; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta S_{ML} \leq S_a\left(N_{\Delta S}\right) \tag{5.23}$$

5.5.3 Fatigue Assessment – Elastic Stress Analysis and Equivalent Stresses

5.5.3.1 Overview

a) An effective total equivalent stress amplitude is used to evaluate the fatigue damage for results obtained from a linear elastic stress analysis. The controlling stress for the fatigue evaluation is the effective total equivalent stress amplitude, defined as one-half of the effective total equivalent stress range $\left(P_L + P_b + Q + F\right)$ calculated for each cycle in the loading histogram.

b) The primary plus secondary plus peak equivalent stress (see Figure 5.1) is the equivalent stress, derived from the highest value across the thickness of a section, of the combination of all primary, secondary,

and peak stresses produced by specified operating pressures and other mechanical loads and by general and local thermal effects and including the effects of gross and local structural discontinuities. Examples of load case combinations for this stress category for typical pressure vessel components are shown in Table 5.3.

5.5.3.2 Assessment Procedure

The following procedure can be used to evaluate protection against failure due to cyclic loading based on the effective total equivalent stress amplitude.

a) STEP 1 – Determine a load history based on the information in the User's Design Specification and the methods in Annex 5.B. The load history should include all significant operating loads and events that are applied to the component. If the exact sequence of loads is not known, alternatives should be examined to establish the most severe fatigue damage, see STEP 6.

b) STEP 2 – For a location in the component subject to a fatigue evaluation, determine the individual stress-strain cycles using the cycle counting methods in Annex 5.B. Define the total number of cyclic stress ranges in the histogram as M .

c) STEP 3 – Determine the equivalent stress range for the k^{th} cycle counted in STEP 2.

1) If the effective alternating equivalent stress is computed using Equation (5.30), then determine the stress tensor at the start and end points (time points ${}^{m}t$ and ${}^{n}t$, respectively) for the k^{th} cycle counted in STEP 2, Determine the local thermal stress at time points ${}^{m}t$ and ${}^{n}t$, ${}^{m}\sigma_{ij,k}^{LT}$ and ${}^{n}\sigma_{ij,k}^{LT}$, respectively, as described in Annex 5.C. The component stress ranges between time points ${}^{m}t$ and ${}^{n}t$ and the effective equivalent stress ranges for use in Equation (5.30) are calculated using Equations (5.24) through (5.27).

$$\Delta\sigma_{ij,k} = \left({}^{m}\sigma_{ij,k} - {}^{m}\sigma_{ij,k}^{LT}\right) - \left({}^{n}\sigma_{ij,k} - {}^{n}\sigma_{ij,k}^{LT}\right) \tag{5.24}$$

$$\left(\Delta S_{P,k} - \Delta S_{LT,k}\right) = \frac{1}{\sqrt{2}}\begin{bmatrix} \left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \\ \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right) \end{bmatrix}^{0.5} \tag{5.25}$$

$$\Delta\sigma_{ij,k}^{LT} = {}^{m}\sigma_{ij,k}^{LT} - {}^{n}\sigma_{ij,k}^{LT} \tag{5.26}$$

$$\Delta S_{LT,k} = \frac{1}{\sqrt{2}}\left[\left(\Delta\sigma_{11,k}^{LT} - \Delta\sigma_{22,k}^{LT}\right)^2 + \left(\Delta\sigma_{11,k}^{LT} - \Delta\sigma_{33,k}^{LT}\right)^2 + \left(\Delta\sigma_{22,k}^{LT} - \Delta\sigma_{33,k}^{LT}\right)^2\right]^{0.5} \tag{5.27}$$

2) If the effective alternating equivalent stress is computed using Equation (5.36), then determine the stress tensor at the start and end points (time points ${}^{m}t$ and ${}^{n}t$, respectively) for the k^{th} cycle counted in STEP 2, The component stress ranges between time points ${}^{m}t$ and ${}^{n}t$, and the effective equivalent stress range for use in Equation (5.36) are given by Equations (5.28) and (5.29) , respectively.

$$\Delta\sigma_{ij,k} = {}^{m}\sigma_{ij,k} - {}^{n}\sigma_{ij,k} \tag{5.28}$$

$$\Delta S_{P,k} = \frac{1}{\sqrt{2}}\left[\begin{array}{l}\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \\ \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right)\end{array}\right]^{0.5} \tag{5.29}$$

d) STEP 4 – Determine the effective alternating equivalent stress amplitude for the k^{th} cycle using the results from STEP 3..

$$S_{alt,k} = \frac{K_f \cdot K_{e,k} \cdot \left(\Delta S_{P,k} - \Delta S_{LT,k}\right) + K_{v,k} \cdot \Delta S_{LT,k}}{2} \tag{5.30}$$

1) If the local notch or effect of the weld is accounted for in the numerical model, then $K_f = 1.0$ in Equations (5.30) and (5.36). If the local notch or effect of the weld is not accounted for in the numerical model, then a fatigue strength reduction factor, K_f, shall be included. Recommended values for fatigue strength reduction factors for welds are provided in Tables 5.11 and 5.12.

2) The fatigue penalty factor, $K_{e,k}$, in Equations (5.30) and (5.36) is evaluated using the following equations where the parameters m and n are determined from Table 5.13 and S_{PS} and $\Delta S_{n,k}$ are defined in paragraph 5.5.6.1. For $K_{e,k}$ values greater than 1.0, the simplified elastic-plastic criteria of paragraph 5.5.6.2 shall be satisfied.

$$K_{e,k} = 1.0 \qquad \textit{for} \quad \Delta S_{n,k} \le S_{PS} \tag{5.31}$$

$$K_{e,k} = 1.0 + \frac{(1-n)}{n(m-1)}\left(\frac{\Delta S_{n,k}}{S_{PS}} - 1\right) \qquad \textit{for} \quad S_{PS} < \Delta S_{n,k} < mS_{PS} \tag{5.32}$$

$$K_{e,k} = \frac{1}{n} \qquad \textit{for} \quad \Delta S_{n,k} \ge mS_{PS} \tag{5.33}$$

3) The Poisson correction factor, $K_{v,k}$ in Equation (5.30) is computed using Equation (5.34).

$$K_{v,k} = \left(\frac{1-\nu_e}{1-\nu_p}\right) \tag{5.34}$$

where

$$\nu_p = max\left[0.5 - 0.2\left(\frac{S_{y,k}}{S_{a,k}}\right), \nu_e\right] \tag{5.35}$$

4) The Poisson correction factor, $K_{v,k}$, in Equation (5.34) need not be used if the fatigue penalty factor, $K_{e,k}$, is used for the entire stress range (including $\Delta S_{LT,k}$). In this case, Equation (5.30) becomes:

$$S_{alt,k} = \frac{K_f \cdot K_{e,k} \cdot \Delta S_{P,k}}{2} \tag{5.36}$$

e) STEP 5 – Determine the permissible number of cycles, N_k, for the alternating equivalent stress computed in STEP 4. Fatigue curves based on the materials of construction are provided in Annex 3.F, paragraph 3.F.1.

f) STEP 6 – Determine the fatigue damage for the k^{th} cycle, where the actual number of repetitions of the k^{th} cycle is n_k.

$$D_{f,k} = \frac{n_k}{N_k} \tag{5.37}$$

g) STEP 7 – Repeat STEPs 3 through 6 for all stress ranges, M, identified in the cycle counting process in STEP 2.

h) STEP 8 – Compute the accumulated fatigue damage using the following equation. The location in the component is acceptable for continued operation if this equation is satisfied.

$$D_f = \sum_{k=1}^{M} D_{f,k} \le 1.0 \tag{5.38}$$

i) STEP 9 – Repeat STEPs 2 through 8 for each point in the component subject to a fatigue evaluation.

5.5.3.3 In STEP 4 of paragraph 5.5.3.2, $K_{e,k}$ may be calculated using one of the following methods.

a) Method 1 – The equivalent total strain range from elastic-plastic analysis and the equivalent total strain range from elastic analysis for the point of interest as given below.

$$K_{e,k} = \frac{\left(\Delta\varepsilon_{t,k}\right)_{ep}}{\left(\Delta\varepsilon_{t,k}\right)_{e}} \tag{5.39}$$

where,

$$\left(\Delta\varepsilon_{t,k}\right)_{ep} = \frac{\sqrt{2}}{3}\begin{bmatrix} \left(\Delta e_{11,k} - \Delta e_{22,k}\right)^2 + \left(\Delta e_{22,k} - \Delta e_{33,k}\right)^2 + \\ \left(\Delta e_{33,k} - \Delta e_{11,k}\right)^2 + 1.5\left(\Delta e_{12,k}^2 + \Delta e_{23,k}^2 + \Delta e_{31,k}^2\right) \end{bmatrix}^{0.5} \tag{5.40}$$

$$\left(\Delta\varepsilon_{t,k}\right)_{e} = \frac{\Delta S_{P,k}}{E_{ya,k}} \tag{5.41}$$

The stress range $\Delta S_{P,k}$ is given by Equation (5.29).

b) Method 2 – The alternate plasticity adjustment factors and alternating equivalent stress may be computed using Annex 5.C.

5.5.3.4 In lieu of a detailed stress analysis, stress indices may be used to determine peak stresses around a nozzle opening in accordance with Annex 5.D.

5.5.4 Fatigue Assessment – Elastic-Plastic Stress Analysis and Equivalent Strains

5.5.4.1 Overview

a) The Effective Strain Range is used to evaluate the fatigue damage for results obtained from an elastic-plastic stress analysis. The Effective Strain Range is calculated for each cycle in the loading histogram using either cycle-by-cycle analysis or the Twice Yield Method. For the cycle-by-cycle analysis, a cyclic plasticity algorithm with kinematic hardening shall be used.

b) Twice Yield Method is an elastic-plastic stress analysis performed in a single loading step, based on a specified stabilized cyclic stress range-strain range curve and a specified load range representing a cycle. Stress and strain ranges are the direct output from this analysis. This method is performed in the same manner as a monotonic analysis and does not require cycle-by-cycle analysis of unloading and reloading. The Twice Yield Method can be used with an analysis program without cyclic plasticity capability.

c) For the calculation of the stress range and strain range of a cycle at a point in the component, a stabilized cyclic stress-strain curve and other material properties shall be used based on the average temperature of the cycle being evaluated for each material of construction. The cyclic curve may be that obtained by test for the material, or that which is known to have more conservative cyclic behavior to the material that is specified. Cyclic stress-strain curves are also provided in paragraph 3.D.4 of Annex 3.D for certain materials and temperatures. Other cyclic stress-strain curves may be used that are known to be either more accurate for the application or lead to more conservative results.

5.5.4.2 Assessment Procedure

The following procedure can be used to evaluate protection against failure due to cyclic loading using elastic-plastic stress analysis.

a) STEP 1 – Determine a load history based on the information in the User's Design Specification and the methods in Annex 5.B. The load history should include all significant operating loads and events that are applied to the component.

b) STEP 2 – For a location in the component subject to a fatigue evaluation, determine the individual stress-strain cycles using the cycle counting methods in Annex 5.B. Define the total number of cyclic stress ranges in the histogram as M.

c) STEP 3 – Determine the loadings at the start and end points of the k^{th} cycle counted in STEP 2. Using these data, determine the loading ranges (differences between the loadings at the start and end points of the cycle).

d) STEP 4 – Perform elastic-plastic stress analysis for the k^{th} cycle. For cycle-by-cycle analysis, constant-amplitude loading is cycled using cyclic stress amplitude-strain amplitude curve (paragraph 5.5.4.1). For the Twice Yield Method, the loading at the start point of the cycle is zero and the loading at the end point is the loading range determined in STEP 3. The cyclic stress range-strain range curve is used (paragraph 5.5.4.1). For thermal loading, the loading range in Twice-Yield Method may be applied by specifying the temperature field at the start point for the cycle as an initial condition, and applying the temperature field at the end point for the cycle in a single loading step.

e) STEP 5 – Calculate the Effective Strain Range for the k^{th} cycle.

$$\Delta\varepsilon_{eff,k} = \frac{\Delta S_{P,k}}{E_{ya,k}} + \Delta\varepsilon_{peq,k} \tag{5.42}$$

where, the stress range $\Delta S_{P,k}$ is given by Equation (5.29)

$$\Delta\varepsilon_{peq,k} = \frac{\sqrt{2}}{3}\left[\begin{array}{l} \left(\Delta p_{11,k} - \Delta p_{22,k}\right)^2 + \left(\Delta p_{22,k} - \Delta p_{33,k}\right)^2 + \\ \left(\Delta p_{33,k} - \Delta p_{11,k}\right)^2 + 1.5\left(\Delta p_{12,k}^2 + \Delta p_{23,k}^2 + \Delta p_{31,k}^2\right) \end{array}\right]^{0.5} \tag{5.43}$$

The component stress and plastic strain ranges (differences between the components at the start and end points of the cycle) for the k^{th} cycle are designated as $\Delta\sigma_{ij,k}$ and $\Delta p_{ij,k}$, respectively. However, since a range of loading is applied in a single load step with the Twice Yield Method, the calculated maximum equivalent plastic strain range, $\Delta\varepsilon_{peq,k}$ and the von Mises equivalent stress range $\Delta S_{P,k}$ defined above are typical output variables that can be obtained directly from a stress analysis.

f) STEP 6 – Determine the effective alternating equivalent stress for the k^{th} cycle.

$$S_{alt,k} = \frac{E_{yf} \cdot \Delta\varepsilon_{eff,k}}{2} \tag{5.44}$$

g) STEP 7 – Determine the permissible number of cycles, N_k, for the alternating equivalent stress computed in STEP 6. Fatigue curves based on the materials of construction are provided in Annex 3.F, paragraph 3.F.1.

h) STEP 8 – Determine the fatigue damage for the k^{th} cycle, where the actual number of repetitions of the k^{th} cycle is n_k.

$$D_{f,k} = \frac{n_k}{N_k} \tag{5.45}$$

i) STEP 9 – Repeat STEPs 3 through 8 for all stress ranges, M, identified in the cycle counting process in STEP 2.

j) STEP 10 – Compute the accumulated fatigue damage using the following equation. The location in the component is acceptable for continued operation if this equation is satisfied.

$$\sum_{k=1}^{M} D_{f,k} \le 1.0 \tag{5.46}$$

k) STEP 11 – Repeat STEPs 2 through 10 for each point in the component subject to a fatigue evaluation.

5.5.5 Fatigue Assessment of Welds – Elastic Analysis and Structural Stress

5.5.5.1 Overview

a) An equivalent structural stress range parameter is used to evaluate the fatigue damage for results obtained from a linear elastic stress analysis. The controlling stress for the fatigue evaluation is the structural stress that is a function of the membrane and bending stresses normal to the hypothetical crack plane. This method is recommended for evaluation of welded joints that have not been machined to a smooth profile. Weld joints with controlled smooth profiles may be evaluated using paragraphs 5.5.3 or 5.5.4.

b) Fatigue cracks at pressure vessel welds are typically located at the toe of a weld. For as-welded and weld joints subject to post weld heat treatment, the expected orientation of a fatigue crack is along the weld toe in the through-thickness direction, and the structural stress normal to the expected crack is the stress measure used to correlate fatigue life data. For fillet welded components, fatigue cracking may occur at the toe of the fillet weld or the weld throat, and both locations shall be considered in the

assessment. It is difficult to accurately predict fatigue life at the weld throat due to variability in throat dimension, which is a function of the depth of the weld penetration. It is recommended to perform sensitivity analysis where the weld throat dimension is varied.

c) This fatigue method may only be used when approved by the owner/user.

5.5.5.2 Assessment Procedure

The following procedure can be used to evaluate protection against failure due to cyclic loading using the equivalent structural stress range.

a) STEP 1 – Determine a load history based on the information in the User's Design Specification and the histogram development methods in Annex 5.B. The load history should include all significant operating loads and events that are applied to the component.

b) STEP 2 – For a location at a weld joint subject to a fatigue evaluation, determine the individual stress-strain cycles using the cycle counting methods in Annex 5.B. Define the total number of cyclic stress ranges in the histogram as M.

c) STEP 3 – Determine the elastically calculated membrane and bending stress normal to the assumed hypothetical crack plane at the start and end points (time points ${}^{m}t$ and ${}^{n}t$, respectively) for the k^{th} cycle counted in STEP 2. Using this data, calculate the membrane and bending stress ranges between time points ${}^{m}t$ and ${}^{n}t$, and the maximum, minimum and mean stress.

$$\Delta\sigma^{e}_{m,k} = {}^{m}\sigma^{e}_{m,k} - {}^{n}\sigma^{e}_{m,k} \tag{5.47}$$

$$\Delta\sigma^{e}_{b,k} = {}^{m}\sigma^{e}_{b,k} - {}^{n}\sigma^{e}_{b,k} \tag{5.48}$$

$$\sigma_{max,k} = \max\left[\left({}^{m}\sigma^{e}_{m,k} + {}^{m}\sigma^{e}_{b,k}\right),\ \left({}^{n}\sigma^{e}_{m,k} + {}^{n}\sigma^{e}_{b,k}\right)\right] \tag{5.49}$$

$$\sigma_{min,k} = \min\left[\left({}^{m}\sigma^{e}_{m,k} + {}^{m}\sigma^{e}_{b,k}\right),\ \left({}^{n}\sigma^{e}_{m,k} + {}^{n}\sigma^{e}_{b,k}\right)\right] \tag{5.50}$$

$$\sigma_{mean,k} = \frac{\sigma_{max,k} + \sigma_{min,k}}{2} \tag{5.51}$$

d) STEP 4 – Determine the elastically calculated structural stress range for the k^{th} cycle, $\Delta\sigma^{e}_{k}$, using Equation (5.52).

$$\Delta\sigma^{e}_{k} = \Delta\sigma^{e}_{m,k} + \Delta\sigma^{e}_{b,k} \tag{5.52}$$

e) STEP 5 – Determine the elastically calculated structural strain, $\Delta\varepsilon^{e}_{k}$, from the elastically calculated structural stress, $\Delta\sigma^{e}_{k}$, using Equation (5.53)

$$\Delta\varepsilon^{e}_{k} = \frac{\Delta\sigma^{e}_{k}}{E_{ya,k}} \tag{5.53}$$

The corresponding local nonlinear structural stress and strain ranges, $\Delta\sigma_{k}$ and $\Delta\varepsilon_{k}$, respectively, are determined by simultaneously solving Neuber's Rule, Equation (5.54), and a model for the material hysteresis loop stress-strain curve given by Equation (5.55), see Annex 3.D, paragraph 3.D.4.

$$\Delta\sigma_k \cdot \Delta\varepsilon_k = \Delta\sigma_k^e \cdot \Delta\varepsilon_k^e \tag{5.54}$$

$$\Delta\varepsilon_k = \frac{\Delta\sigma_k}{E_{ya,k}} + 2\left(\frac{\Delta\sigma_k}{2K_{css}}\right)^{\frac{1}{n_{css}}} \tag{5.55}$$

The structural stress range computed solving Equations (5.54) and (5.55) is subsequently modified for low-cycle fatigue using Equation(5.56).

$$\Delta\sigma_k = \left(\frac{E_{ya,k}}{1-\nu^2}\right)\Delta\varepsilon_k \tag{5.56}$$

NOTE: The modification for low-cycle fatigue should always be performed because the exact distinction between high-cycle fatigue and low-cycle fatigue cannot be determined without evaluating the effects of plasticity which is a function of the applied stress range and cyclic stress-strain curve. For high cycle fatigue applications, this procedure will provide correct results, i.e. the elastically calculated structural stress will not be modified.

f) STEP 6 – Compute the equivalent structural stress range parameter for the k^{th} cycle using the following equations. In Equation (5.57), for SI Units, the thickness, t, stress range, $\Delta\sigma_k$, and the equivalent structural stress range parameter, $\Delta S_{ess,k}$, are in mm, MPa, and $MPa/(mm)^{(2-m_{ss})/2m_{ss}}$, respectively, and for US Customary Units, the thickness, t, stress range, $\Delta\sigma_k$, and the equivalent structural stress range parameter, $\Delta S_{ess,k}$, are in $inches$, ksi, and $ksi/(inches)^{(2-m_{ss})/2m_{ss}}$, respectively.

$$\Delta S_{ess,k} = \frac{\Delta\sigma_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \cdot I^{\frac{1}{m_{ss}}} \cdot f_{M,k}} \tag{5.57}$$

where,

$$m_{ss} = 3.6 \tag{5.58}$$

$$t_{ess} = 16\ mm\ (0.625\ in.) \qquad for \qquad t \le 16\ mm\ (0.625\ in.) \tag{5.59}$$

$$t_{ess} = t \qquad for \qquad 16\ mm\ (0.625\ in.) < t < 150\ mm\ (6\ in.) \tag{5.60}$$

$$t_{ess} = 150\ mm\ (6\ in.) \qquad for \qquad t \ge 150\ mm\ (6\ in.) \tag{5.61}$$

$$I^{\frac{1}{m_{ss}}} = \frac{1.23 - 0.364R_{b,k} - 0.17R_{b,k}^2}{1.007 - 0.306R_{b,k} - 0.178R_{b,k}^2} \tag{5.62}$$

$$R_{b,k} = \frac{\left|\Delta\sigma_{b,k}^e\right|}{\left|\Delta\sigma_{m,k}^e\right| + \left|\Delta\sigma_{b,k}^e\right|} \tag{5.63}$$

$$f_{M,k} = \left(1 - R_k\right)^{\frac{1}{m_{ss}}} \qquad for \quad \begin{vmatrix} \sigma_{mean,k} \geq 0.5S_{y,k}, \ and \\ R_k > 0, \ and \\ \left|\Delta\sigma^e_{m,k} + \Delta\sigma^e_{b,k}\right| \leq 2S_{y,k} \end{vmatrix} \tag{5.64}$$

$$f_{M,k} = 1.0 \qquad for \quad \begin{vmatrix} \sigma_{mean,k} < 0.5S_{y,k}, \ or \\ R_k \leq 0, \ or \\ \left|\Delta\sigma^e_{m,k} + \Delta\sigma^e_{b,k}\right| > 2S_{y,k} \end{vmatrix} \tag{5.65}$$

$$R_k = \frac{\sigma_{min,k}}{\sigma_{max,k}} \tag{5.66}$$

g) STEP 7 – Determine the permissible number of cycles, N_k, based on the equivalent structural stress range parameter for the k^{th} cycle computed in STEP 6. Fatigue curves for welded joints are provided in Annex 3.F, paragraph 3.F.2.

h) STEP 8 – Determine the fatigue damage for the k^{th} cycle, where the actual number of repetitions of the k^{th} cycle is n_k.

$$D_{f,k} = \frac{n_k}{N_k} \tag{5.67}$$

i) STEP 9 – Repeat STEPs 6 through 8 for all stress ranges, M, identified in the cycle counting process in STEP 3.

j) STEP 10 – Compute the accumulated fatigue damage using the following equation. The location along the weld joint is suitable for continued operation if this equation is satisfied.

$$D_f = \sum_{i=1}^{M} D_{f,k} \leq 1.0 \tag{5.68}$$

k) STEP 11 – Repeat STEPs 5 through 10 for each point along the weld joint that is subject to a fatigue evaluation.

5.5.5.3 Assessment Procedure Modifications

The assessment procedure in paragraph 5.5.5.2 may be modified as shown below.

a) Multiaxial Fatigue – If the structural shear stress range is not negligible, i.e. $\Delta\tau_k > \Delta\sigma_k / 3$, a modification should be made when computing the equivalent structural stress range. Two conditions need to be considered:

1) If $\Delta\sigma_k$ and $\Delta\tau_k$ are out of phase, the equivalent structural stress range $\Delta S_{ess,k}$ in Equation (5.57) should be replaced by:

$$\Delta S_{ess,k} = \frac{1}{F(\delta)}\left[\left(\frac{\Delta\sigma_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \cdot I^{\frac{1}{m_{ss}}} \cdot f_{M,k}}\right)^2 + 3\left(\frac{\Delta\tau_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \cdot I_\tau^{\frac{1}{m_{ss}}}}\right)^2\right]^{0.5} \tag{5.69}$$

where

$$I_\tau^{\frac{1}{m_{ss}}} = \frac{1.23 - 0.364R_{b\tau,k} - 0.17R_{b\tau,k}^2}{1.007 - 0.306R_{b\tau,k} - 0.178R_{b\tau,k}^2} \tag{5.70}$$

$$R_{b\tau,k} = \frac{\left|\Delta\tau_{b,k}^e\right|}{\left|\Delta\tau_{m,k}^e\right| + \left|\Delta\tau_{b,k}^e\right|} \tag{5.71}$$

$$\Delta\tau_k = \Delta\tau_{m,k}^e + \Delta\tau_{b,k}^e \tag{5.72}$$

$$\Delta\tau_{m,k}^e = {}^m\tau_{m,k}^e - {}^n\tau_{m,k}^e \tag{5.73}$$

$$\Delta\tau_{b,k}^e = {}^m\tau_{b,k}^e - {}^n\tau_{b,k}^e \tag{5.74}$$

In Equation (5.69), $F(\delta)$ is a function of the out-of-phase angle between $\Delta\sigma_k$ and $\Delta\tau_k$ if both loading modes can be described by sinusoidal functions, or:

$$F(\delta) = \frac{1}{\sqrt{2}}\left[1 + \left[1 - \frac{12\cdot\Delta\sigma_k^2\cdot\Delta\tau_k^2\cdot\sin^2[\delta]}{\left[\Delta\sigma_k^2 + 3\Delta\tau_k^2\right]^2}\right]^{0.5}\right]^{0.5} \tag{5.75}$$

A conservative approach is to ignore the out-of-phase angle and recognize the existence of a minimum possible value for $F(\delta)$ in Equation (5.75) given by:

$$F(\delta) = \frac{1}{\sqrt{2}} \tag{5.76}$$

2) If $\Delta\sigma_k$ and $\Delta\tau_k$ are in-phase the equivalent structural stress range $\Delta S_{ess,k}$ is given by Equation (5.69) with $F(\delta) = 1.0$.

b) Weld Quality – If a defect exists at the toe of a weld that can be characterized as a crack-like flaw, i.e. undercut, and this defect exceeds the value permitted by Part 7, then a reduction in fatigue life shall calculated by substituting the value of $I^{1/m_{ss}}$ in Equation (5.62) with the value given by Equation (5.77). In this equation, a is the depth of the crack-like flaw at the weld toe. Equation (5.77) is valid only when $a/t \le 0.1$.

$$I^{\frac{1}{m_{ss}}} = \frac{1.229 - 0.365R_{b,k} + 0.789\left(\frac{a}{t}\right) - 0.17R_{b,k}^2 + 13.771\left(\frac{a}{t}\right)^2 + 1.243R_{b,k}\left(\frac{a}{t}\right)}{1 - 0.302R_{b,k} + 7.115\left(\frac{a}{t}\right) - 0.178R_{b,k}^2 + 12.903\left(\frac{a}{t}\right)^2 - 4.091R_{b,k}\left(\frac{a}{t}\right)} \tag{5.77}$$

5.5.6 Ratcheting Assessment – Elastic Stress Analysis

5.5.6.1 Elastic Ratcheting Analysis Method

a) To evaluate protection against ratcheting the following limit shall be satisfied.

$$\Delta S_{n,k} \le S_{PS} \tag{5.78}$$

b) The primary plus secondary equivalent stress range, $\Delta S_{n,k}$, is the equivalent stress range, derived from the highest value across the thickness of a section, of the combination of linearized general or local primary membrane stresses plus primary bending stresses plus secondary stresses $\left(P_L + P_b + Q\right)$, produced by specified operating pressure and other specified mechanical loads and by general thermal effects. The effects of gross structural discontinuities but not of local structural discontinuities (stress concentrations) shall be included. Examples of this stress category for typical pressure vessel components are shown in Table 5.6.

c) The maximum range of this equivalent stress is limited to S_{PS}. The quantity S_{PS} represents a limit on the primary plus secondary equivalent stress range and is defined in paragraph 5.5.6.1.d. In the determination of the maximum primary plus secondary equivalent stress range, it may be necessary to consider the effects of multiple cycles where the total stress range may be greater than the stress range of any of the individual cycles. In this case, the value of S_{PS} may vary with the specified cycle, or combination of cycles, being considered since the temperature extremes may be different in each case. Therefore, care shall be exercised to assure that the applicable value of S_{PS} for each cycle, or combination of cycles, is used (see paragraph 5.5.3).

d) The allowable limit on the primary plus secondary stress range, S_{PS}, is computed as the larger of the quantities shown below.

   1) Three times the average of the S values for the material from Annex 3.A at the highest and lowest temperatures during the operational cycle.

   2) Two times the average of the S_y values for the material from Annex 3.D at the highest and lowest temperatures during the operational cycle, except that the value from paragraph 5.5.6.1.d.1 shall be used when the ratio of the minimum specified yield strength to ultimate tensile strength exceeds 0.70 or the value of S is governed by time-dependent properties as indicated in Annex 3.A.

5.5.6.2 Simplified Elastic-Plastic Analysis

The equivalent stress limit on the range of primary plus secondary equivalent stress in paragraph 5.5.6.1 may be exceeded provided all of the following are true:

a) The range of primary plus secondary membrane plus bending equivalent stress, excluding thermal bending stress, is less than S_{PS}.

b) The value of the alternating stress range in paragraph 5.5.3.2.d is multiplied by the factor $K_{e,k}$ (see Equations (5.31) through (5.33), or paragraph 5.5.3.3) .

c) The material of the component has a ratio of the specified minimum yield strength to specified minimum tensile strength of less than or equal to 0.80.

d) The component meets the secondary equivalent stress range requirements of paragraph 5.5.6.3.

5.5.6.3 Thermal Stress Ratcheting Assessment

The allowable limit on the secondary equivalent stress range from thermal loading to prevent ratcheting, when applied in conjunction with a steady state general or local primary membrane equivalent stress, is determined below. This procedure can only be used with an assumed linear or parabolic distribution of a secondary stress range (e.g. thermal stress).

a) STEP 1 – Determine the ratio of the primary membrane stress to the specified minimum yield strength from Annex 3.D, at the average temperature of the cycle.

$$X = \left(\frac{P_m}{S_y}\right) \tag{5.79}$$

b) STEP 2 – Compute the secondary equivalent stress range from thermal loading, ΔQ using elastic analysis methods.

c) STEP 3 – Determine the allowable limit on the secondary equivalent stress range from thermal loading, S_Q.

1. For a secondary equivalent stress range from thermal loading with a linear variation through the wall thickness:

$$S_Q = S_y\left(\frac{1}{X}\right) \qquad for \qquad 0 < X < 0.5 \tag{5.80}$$

$$S_Q = 4.0S_y\left(1 - X\right) \qquad for \qquad 0.5 \le X \le 1.0 \tag{5.81}$$

2. For a secondary equivalent stress range from thermal loading with a parabolic constantly increasing or decreasing variation through the wall thickness:

$$S_Q = S_y\left(\frac{1}{0.1224 + 0.9944X^2}\right) \qquad for \qquad 0.0 < X < 0.615 \tag{5.82}$$

$$S_Q = 5.2S_y\left(1 - X\right) \qquad for \qquad 0.615 \le X \le 1.0 \tag{5.83}$$

d) STEP 4 – To demonstrate protection against ratcheting, the following criteria shall be satisfied.

$$\Delta Q \le S_Q \tag{5.84}$$

5.5.6.4 Progressive Distortion of Non-Integral Connections

Screwed-on caps, screwed-in plugs, shear ring closures, and breech lock closures are examples of non-integral connections that are subject to failure by bell-mouthing or other types of progressive deformation. If any combination of applied loads produces yielding, such joints are subject to ratcheting because the mating members may become loose at the end of each complete operating cycle and may start the next cycle in a new relationship with each other, with or without manual manipulation. Additional distortion may occur in each cycle so that interlocking parts, such as threads, can eventually lose engagement. Therefore primary plus secondary equivalent stresses that produce slippage between the parts of a non-integral connection in

which disengagement could occur as a result of progressive distortion, shall be limited to the minimum specified yield strength at temperature, S_y, or evaluated using the procedure in paragraph 5.5.7.2.

5.5.7 Ratcheting Assessment – Elastic-Plastic Stress Analysis

5.5.7.1 Overview

To evaluate protection against ratcheting using elastic-plastic analysis, an assessment is performed by application, removal and re-application of the applied loadings. If protection against ratcheting is satisfied, it may be assumed that progression of the stress-strain hysteresis loop along the strain axis cannot be sustained with cycles and that the hysteresis loop will stabilize. A separate check for plastic shakedown to alternating plasticity is not required. The following assessment procedure can be used to evaluate protection against ratcheting using elastic-plastic analysis.

5.5.7.2 Assessment Procedure

a) STEP 1 – Develop a numerical model of the component including all relevant geometry characteristics. The model used for analysis shall be selected to accurately represent the component geometry, boundary conditions, and applied loads.

b) STEP 2 – Define all relevant loads and applicable load cases (see Table 5.1).

c) STEP 3 – An elastic-perfectly plastic material model shall be used in the analysis. The von Mises yield function and associated flow rule should be utilized. The yield strength defining the plastic limit shall be the minimum specified yield strength at temperature from Annex 3.D. The effects of non-linear geometry shall be considered in the analysis.

d) STEP 4 – Perform an elastic-plastic analysis for the applicable loading from STEP 2 for a number of repetitions of a loading event (see Annex 5.B), or, if more than one event is applied, of two events that are selected so as to produce the highest likelihood of ratcheting.

e) STEP 5 – The ratcheting criteria below shall be evaluated after application of a minimum of three complete repetitions of the cycle. Additional cycles may need to be applied to demonstrate convergence. If any one of the following conditions is met, the ratcheting criteria are satisfied. If the criteria shown below are not satisfied, the component configuration (i.e. thickness) shall be modified or applied loads reduced and the analysis repeated.

    1) There is no plastic action (i.e. zero plastic strains incurred) in the component.

    2) There is an elastic core in the primary-load-bearing boundary of the component.

    3) There is not a permanent change in the overall dimensions of the component. This can be demonstrated by developing a plot of relevant component dimensions versus time between the last and the next to the last cycles.

5.6 Supplemental Requirements for Stress Classification in Nozzle Necks

The following classification of stresses shall be used for stress in a nozzle neck. The classification of stress in the shell shall be in accordance with paragraph 5.2.2.2.

a) Within the limits of reinforcement given by paragraph 4.5, whether or not nozzle reinforcement is provided, the following classification shall be applied.

    1) A P_m classification is applicable to equivalent stresses resulting from pressure induced general membrane stresses as well as stresses, other than discontinuity stresses, due to external loads and moments including those attributable to restrained free end displacements of the attached pipe.

    2) A P_L classification shall be applied to local primary membrane equivalent stresses derived from discontinuity effects plus primary bending equivalent stresses due to combined pressure and

external loads and moments including those attributable to restrained free end displacements of the attached pipe.

3) A $P_L + P_b + Q$ classification (see paragraph 5.5.2) shall apply to primary plus secondary equivalent stresses resulting from a combination of pressure, temperature, and external loads and moments, including those due to restrained free end displacements of the attached pipe.

b) Outside of the limits of reinforcement given in paragraph 4.5, the following classification shall be applied.

1) A P_m classification is applicable to equivalent stresses resulting from pressure induced general membrane stresses as well as the average stress across the nozzle thickness due to externally applied nozzle axial, shear, and torsional loads other than those attributable to restrained free end displacement of the attached pipe.

2) A $P_L + P_b$ classification is applicable to the equivalent stresses resulting from adding those stresses classified as P_m to those due to externally applied bending moments except those attributable to restrained free end displacement of the pipe.

3) A $P_L + P_b + Q$ classification (see paragraph 5.5.2) is applicable to equivalent stresses resulting from all pressure, temperature, and external loads and moments, including those attributable to restrained free end displacements of the attached pipe.

c) Beyond the limits of reinforcement, the S_{PS} limit on the range of primary plus secondary equivalent stress may be exceeded as provided in paragraph 5.5.6.2, except that in the evaluation of the range of primary plus secondary equivalent stress, $P_L + P_b + Q$, stresses resulting from the restrained free end displacements of the attached pipe may also be excluded. The range of membrane plus bending equivalent stress attributable solely to the restrained free end displacements of the attached piping shall be less than S_{PS}.

5.7 Supplemental Requirements for Bolts

5.7.1 Design Requirements

a) The number and cross-sectional area of bolts required to resist the design pressure shall be determined in accordance with the procedures of paragraph 4.16. The allowable bolt stress shall be obtained from Part 3.

b) When sealing is effected by a seal weld instead of a gasket, the gasket factor, m, and the minimum gasket seating stress, y, may be taken as zero.

c) When gaskets are used for pre-service testing only, the design is satisfactory if the above requirements are satisfied for m and y factors equal to zero, and the requirements of paragraphs 5.7.1 and 5.7.2 are satisfied when the appropriate m and y factors are used for the test gasket.

5.7.2 Service Stress Requirements

Actual service stress in bolts, such as those produced by the combination of preload, pressure, and differential expansion, may be higher than the allowable stress values given in Annex 3.A.

a) The maximum value of service stress, averaged across the bolt cross section and neglecting stress concentrations, shall not exceed two times the allowable stress values in paragraph 3.A.2.2 of Annex 3.A.

b) The maximum value of service stress, except as restricted by paragraph 5.7.3.1.b at the periphery of the bolt cross section resulting from direct tension plus bending and neglecting stress concentrations, shall

not exceed three times the allowable stress values in paragraph 3.A.2 of Annex 3.A. When the bolts are tightened by methods other than heaters, stretchers, or other means which minimize residual torsion, the stress measure used in the evaluation shall be the equivalent stress as defined in Equation 5.1.

5.7.3 Fatigue Assessment Of Bolts

5.7.3.1 The suitability of bolts for cyclic operation shall be determined in accordance with the following procedures unless the vessel on which they are installed meets all the conditions of paragraph 5.5.2 (a fatigue analysis is not required).

a) Bolts made of materials which have minimum specified tensile strengths of less than 689 MPa (100,000 psi) shall be evaluated for cyclic operation using the method in paragraph 5.5.3, using the applicable design fatigue curves (see Annex 3.F), and, unless it can be shown by analysis or test that a lower value is appropriate, the fatigue strength reduction factor used in the evaluation shall not be less than 4.0.

b) High strength alloy steel bolts and studs shall be evaluated for cyclic operation using the methodology in paragraph 5.5.3 with the applicable design fatigue curve of Annex 3.F, provided all of the following are true:

   1) The material is one of the following: SA-193 Grade B7 or B16, SA-320 Grade L43, SA-540 Grades B23 and B24, heat treated in accordance with Section 5 of SA-540.

   2) The maximum value of the service stress at the periphery of the bolt cross section (resulting from direct tension plus bending and neglecting stress concentrations) shall not exceed $2.7S$, if the higher of the two fatigue design curves for high strength bolting given in Annex 3.F is used (the $2S$ limit for direct tension is unchanged).

   3) The threads shall be of a V-Type, having a minimum thread root radius no smaller than 0.076 mm (0.003 in).

   4) The fillet radii at the end of the shank shall be such that the ratio of fillet radius to shank diameter is not less than 0.060.

   5) The fatigue strength reduction factor used in the evaluation shall not be less than 4.0.

5.7.3.2 The bolts shall be acceptable for the specified cyclic operation application of loads and thermal stresses provided the fatigue damage fraction, D_f, is less than or equal to 1.0 (see paragraph 5.5.3).

5.8 Supplemental Requirements for Perforated Plates

Perforated plates may be analyzed using any of the procedures in this Part if the holes are explicitly included in the numerical model used for the stress analysis. An elastic stress analysis option utilizing the concept of an effective solid plate is described Annex 5.E.

5.9 Supplemental Requirements for Layered Vessels

The equations developed for solid wall cylindrical shells, spherical shells, or heads as expressed in this Part may be applied to layered cylindrical shells, spherical shells or heads, provided that in-plane shear force on each layer is adequately supported by the weld joint. In addition, consideration shall be given to the construction details in the zones of load application. In order to assure solid wall equivalence for layered cylindrical shells, spherical shells, or heads as described above, all cylindrical shells, spherical shells, or heads subjected to radial forces and/or longitudinal bending moments due to discontinuities or externally applied loads shall have all layers adequately bonded together to resist any longitudinal shearing forces resulting from the radial forces and/or longitudinal bending moments acting on the sections. For example, the use of the girth weld to bond layers together is shown in Figures 5.2, 5.3, and 5.4. The required width of the attachment weld at the midpoint of the weld depth is given by Equation (5.85).

$$w = 1.88\left(\frac{M_o}{t \cdot S}\right) \tag{5.85}$$

In Equation (5.85), the parameter M_o is the longitudinal bending moment per unit length of circumference existing at the weld junction of a layered cylindrical shells, spherical shells, or head. This parameter is determined from a stress analysis considering the pressure loading and all externally applied loads such as M_1, Q_1, and F_1.

5.10 Experimental Stress Analysis

Requirements for determining stresses in parts using experimental stress analysis are provided in Annex 5.F.

5.11 Fracture Mechanic Evaluations

Fracture mechanics evaluations performed to determine the MDMT in accordance with paragraph 3.11.2.8 shall be in accordance with API/ASME FFS-1. Residual stresses resulting from welding shall be considered along with primary and secondary stresses in all fracture mechanics calculations.

5.12 Definitions

1. Bending Stress – The variable component of normal stress, the variation may or may not be linear across the section thickness.
2. Bifurcation Buckling – The point of instability where there is a branch in the primary load versus displacement path for a structure.
3. Event – The User's Design Specification may include one or more events that produce fatigue damage. Each event consists of loading components specified at a number of time points over a time period and is repeated a specified number of times. For example, an event may be the startup, shutdown, upset condition, or any other cyclic action. The sequence of multiple events may be specified or random.
4. Cycle – A cycle is a relationship between stress and strain that is established by the specified loading at a location in a vessel or component. More than one stress-strain cycle may be produced at a location, either within an event or in transition between two events, and the accumulated fatigue damage of the stress-strain cycles determines the adequacy for the specified operation at that location. This determination shall be made with respect to the stabilized stress-strain cycle.
5. Cyclic Loading – A service in which fatigue becomes significant due to the cyclic nature of the mechanical and/or thermal loads. A screening criteria is provided in paragraph 5.5.2 that can be used to determine if a fatigue analysis should be included as part of the vessel design.
6. Fatigue – The conditions leading to fracture under repeated or fluctuating stresses having a maximum value less than the tensile strength of the material.
7. Fatigue Endurance Limit – The maximum stress below which a material can undergo 10^{11} alternating stress cycles without failure.
8. Fatigue Strength Reduction Factor – A stress intensification factor which accounts for the effect of a local structural discontinuity (stress concentration) on the fatigue strength. It is the ratio of the fatigue strength of a component without a discontinuity or weld joint to the fatigue strength of that same component with a discontinuity or weld joint. Values for some specific cases are empirically determined (e.g. socket welds). In the absence of experimental data, the stress intensification factor can be developed from a theoretical stress concentration factor derived from the theory of elasticity or based on the guidance provided in Tables 5.11 and 5.12.
9. Fracture Mechanics – an engineering discipline concerned with the behavior of cracks in materials. Fracture mechanics models provide mathematical relationships for critical combinations of stress, crack size and fracture toughness that lead to crack propagation. Linear Elastic Fracture Mechanics (LEFM)

approaches apply to cases where crack propagation occurs during predominately elastic loading with negligible plasticity. Elastic-Plastic Fracture Mechanics (EPFM) methods are suitable for materials that undergo significant plastic deformation during crack propagation.

10. Gross Structural Discontinuity – A source of stress or strain intensification that affects a relatively large portion of a structure and has a significant effect on the overall stress or strain pattern or on the structure as a whole. Examples of gross structural discontinuities are head-to-shell and flange-to-shell junctions, nozzles, and junctions between shells of different diameters or thicknesses.
11. Local Primary Membrane Stress – Cases arise in which a membrane stress produced by pressure, or other mechanical loading associated with a primary and/or a discontinuity effect would, if not limited, produce excessive distortion in the transfer of load to other portions of the structure. Conservatism requires that such a stress be classified as a local primary membrane stress even though it has some characteristics of a secondary stress.
12. Local Structural Discontinuity – A source of stress or strain intensification which affects a relatively small volume of material and does not have a significant effect on the overall stress or strain pattern, or on the structure as a whole. Examples are small fillet radii, small attachments, and partial penetration welds.
13. Membrane Stress – The component of normal stress that is uniformly distributed and equal to the average value of stress across the thickness of the section under consideration.
14. Normal Stress – The component of stress normal to the plane of reference. Usually the distribution of normal stress is not uniform through the thickness of a part.
15. Operational Cycle – An operational cycle is defined as the initiation and establishment of new conditions followed by a return to the conditions that prevailed at the beginning of the cycle. Three types of operational cycles are considered: the startup-shutdown cycle, defined as any cycle which has atmospheric temperature and/or pressure as one of its extremes and normal operating conditions as its other extreme; the initiation of, and recovery from, any emergency or upset condition or pressure test condition that shall be considered in the design; and the normal operating cycle, defined as any cycle between startup and shutdown which is required for the vessel to perform its intended purpose.
16. Peak Stress – The basic characteristic of a peak stress is that it does not cause any noticeable distortion and is objectionable only as a possible source of a fatigue crack or a brittle fracture. A stress that is not highly localized falls into this category if it is of a type that cannot cause noticeable distortion. Examples of peak stress are: the thermal stress in the austenitic steel cladding of a carbon steel vessel, the thermal stress in the wall of a vessel or pipe caused by a rapid change in temperature of the contained fluid, and the stress at a local structural discontinuity.
17. Primary Stress – A normal or shear stress developed by the imposed loading which is necessary to satisfy the laws of equilibrium of external and internal forces and moments. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses which considerably exceed the yield strength will result in failure or at least in gross distortion. A thermal stress is not classified as a primary stress. Primary membrane stress is divided into general and local categories. A general primary membrane stress is one that is distributed in the structure such that no redistribution of load occurs as a result of yielding. Examples of primary stress are general membrane stress in a circular cylindrical or a spherical shell due to internal pressure or to distributed live loads and the bending stress in the central portion of a flat head due to pressure. Cases arise in which a membrane stress produced by pressure or other mechanical loading and associated with a primary and/or a discontinuity effect would, if not limited, produce excessive distortion in the transfer of load to other portions of the structure. Conservatism requires that such a stress be classified as a local primary membrane stress even though it has some characteristics of a secondary stress. Finally a primary bending stress can be defined as a bending stress developed by the imposed loading which is necessary to satisfy the laws of equilibrium of external and internal forces and moments.
18. Ratcheting – A progressive incremental inelastic deformation or strain that can occur in a component subjected to variations of mechanical stress, thermal stress, or both (thermal stress ratcheting is partly or wholly caused by thermal stress). Ratcheting is produced by a sustained load acting over the full cross section of a component, in combination with a strain controlled cyclic load or temperature distribution that is alternately applied and removed. Ratcheting causes cyclic straining of the material, which can result in

failure by fatigue and at the same time produces cyclic incremental growth of a structure, which could ultimately lead to collapse.

19. Secondary Stress – A normal stress or a shear stress developed by the constraint of adjacent parts or by self-constraint of a structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions that cause the stress to occur and failure from one application of the stress is not to be expected. Examples of secondary stress are a general thermal stress and the bending stress at a gross structural discontinuity.
20. Shakedown – Caused by cyclic loads or cyclic temperature distributions which produce plastic deformations in some regions of the component when the loading or temperature distribution is applied, but upon removal of the loading or temperature distribution, only elastic primary and secondary stresses are developed in the component, except in small areas associated with local stress (strain) concentrations. These small areas shall exhibit a stable hysteresis loop, with no indication of progressive deformation. Further loading and unloading, or applications and removals of the temperature distribution shall produce only elastic primary and secondary stresses.
21. Shear Stress – The component of stress tangent to the plane of reference.
22. Stress Concentration Factor – The ratio of the maximum stress to the average section stress or bending stress.
23. Stress Cycle – A stress cycle is a condition in which the alternating stress difference goes from an initial value through an algebraic maximum value and an algebraic minimum value and then returns to the initial value. A single operational cycle may result in one or more stress cycles.
24. Thermal Stress – A self-balancing stress produced by a non-uniform distribution of temperature or by differing thermal coefficients of expansion. Thermal stress is developed in a solid body whenever a volume of material is prevented from assuming the size and shape that it normally should under a change in temperature. For the purpose of establishing allowable stresses, two types of thermal stress are recognized, depending on the volume or area in which distortion takes place. A general thermal stress that is associated with distortion of the structure in which it occurs. If a stress of this type, neglecting stress concentrations, exceeds twice the yield strength of the material, the elastic analysis may be invalid and successive thermal cycles may produce incremental distortion. Therefore this type is classified as a secondary stress. Examples of general thermal stress are: the stress produced by an axial temperature distribution in a cylindrical shell, the stress produced by the temperature difference between a nozzle and the shell to which It is attached, and the equivalent linear stress produced by the radial temperature distribution in a cylindrical shell. A Local thermal stress is associated with almost complete suppression of the differential expansion and thus produces no significant distortion. Such stresses shall be considered only from the fatigue standpoint and are therefore classified as local stresses. Examples of local thermal stresses are the stress in a small hot spot in a vessel wall, the difference between the non-linear portion of a through-wall temperature gradient in a cylindrical shell, and the thermal stress in a cladding material that has a coefficient of expansion different from that of the base metal.

5.13 Nomenclature

a radius of hot spot or heated area within a plate or the depth of a flaw at a weld toe, as applicable.

α thermal expansion coefficient of the material at the mean temperature of two adjacent points, the thermal expansion coefficient of material evaluated at the mean temperature of the cycle, or the cone angle, as applicable.

α_1 thermal expansion coefficient of material 1 evaluated at the mean temperature of the cycle.

α_2 thermal expansion coefficient of material 2 evaluated at the mean temperature of the cycle.

α_{sl} material factor for the multiaxial strain limit.

β_{cr} capacity reduction factor.

C_1 factor for a fatigue analysis screening based on Method B.

C_2 factor for a fatigue analysis screening based on Method B.

D_f	is cumulative fatigue damage.
$D_{f,k}$	is fatigue damage for the k^{th} cycle.
D_{ε}	cumulative strain limit damage.
$D_{\varepsilon form}$	strain limit damage from forming.
$D_{\varepsilon,k}$	strain limit damage for the k^{th} loading condition.
$\Delta e_{ij,k}$	change in total strain range components minus the free thermal strain at the point under evaluation for the k^{th} cycle.
$\Delta\varepsilon_k$	local nonlinear structural strain range at the point under evaluation for the k^{th} cycle.
$\Delta\varepsilon_k^e$	elastically calculated structural strain range at the point under evaluation for the k^{th} cycle.
$\left(\Delta\varepsilon_{t,k}\right)_{ep}$	equivalent strain range for the k^{th} cycle, computed from elastic-plastic analysis, using the total strain less the free thermal strain.
$\left(\Delta\varepsilon_{t,k}\right)_e$	equivalent strain range for the k^{th} cycle, computed from elastic analysis, using the total strain less the free thermal strain.
$\Delta\varepsilon_{ij,k}$	component strain range for the k^{th} cycle, computed using the total strain less the free thermal strain
$\Delta\varepsilon_{peq,k}$	equivalent plastic strain range for the k^{th} loading condition or cycle.
$\Delta\varepsilon_{eff,k}$	Effective Strain Range for the k^{th} cycle.
$\Delta p_{ij,k}$	change in plastic strain range components at the point under evaluation for the k^{th} loading condition or cycle.
ΔP_N	maximum design range of pressure associated with $N_{\Delta P}$.
$\Delta S_{n,k}$	primary plus secondary equivalent stress range.
$\Delta S_{P,k}$	range of primary plus secondary plus peak equivalent stress for the k^{th} cycle.
$\Delta S_{LT,k}$	local thermal equivalent stress for the k^{th} cycle.
$\Delta S_{ess,k}$	equivalent structural stress range parameter for the k^{th} cycle.
ΔS_{ML}	equivalent stress range computed from the specified full range of mechanical loads, excluding pressure but including piping reactions.
ΔQ	range of secondary equivalent stress.
ΔT	operating temperature range.
ΔT_E	effective number of changes in metal temperature between any two adjacent points.
ΔT_M	temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation with $N_{\Delta TM}$.
ΔT_N	temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation with $N_{\Delta TN}$.
ΔT_R	temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation with $N_{\Delta TR}$.
$\Delta\sigma_i$	stress range associated with the principal stress in the i^{th} direction.

$\Delta\sigma_{ij}$ stress tensor range.

$\Delta\sigma_k$ local nonlinear structural stress range at the point under evaluation for the k^{th} cycle.

$\Delta\sigma_k^e$ elastically calculated structural stress range at the point under evaluation for the k^{th} cycle.

$\Delta\sigma_{b,k}^e$ elastically calculated structural bending stress range at the point under evaluation for the k^{th} cycle.

$\Delta\sigma_{ij,k}$ stress tensor range at the point under evaluation for the k^{th} cycle.

$\Delta\sigma_{m,k}^e$ elastically calculated structural membrane stress range at the point under evaluation for the k^{th} cycle.

$\Delta\tau_k$ structural shear stress range at the point under evaluation for the k^{th} cycle.

$\Delta\tau_{b,k}^e$ elastically calculated bending component of the structural shear stress range at the point under evaluation for the k^{th} cycle.

$\Delta\tau_{m,k}^e$ elastically calculated membrane component of the structural shear stress range at the point under evaluation for the k^{th} cycle.

δ out-of-phase angle between $\Delta\sigma_k$ and $\Delta\tau_k$ for the k^{th} cycle.

E_y unmodified Young's modulus for plate material.

E_{yf} value of modulus of elasticity on the fatigue curve being utilized.

$E_{ya,k}$ value of modulus of elasticity of the material at the point under consideration, evaluated at the mean temperature of the k^{th} cycle.

E_{y1} Young’s Modulus of material 1 evaluated at the mean temperature of the cycle.

E_{y2} Young’s Modulus of material 2 evaluated at the mean temperature of the cycle.

E_{ym} Young’s Modulus of the material evaluated at the mean temperature of the cycle.

ε_{cf} cold forming strain.

ε_{Lu} uniaxial strain limit.

$\varepsilon_{L,k}$ limiting triaxial strain.

$f_{M,k}$ mean stress correction factor for the k^{th} cycle.

F additional stress produced by the stress concentration over and above the nominal stress level resulting from operating loadings.

F_1 externally applied axial force.

$F(\delta)$ a fatigue modification factor based on the out-of-phase angle between $\Delta\sigma_k$ and $\Delta\tau_k$.

I correction factor used in the structural stress evaluation.

I_τ correction factor used in the structural shear stress evaluation.

K_{css} material parameter for the cyclic stress-strain curve model.

$K_{e,k}$ fatigue penalty factor for the k^{th} cycle.

$K_{v,k}$ plastic Poisson’s ratio adjustment for local thermal and thermal bending stresses for the k^{th} cycle.

K_f fatigue strength reduction factor used to compute the cyclic stress amplitude or range .

K_L equivalent stress load factor.

K_m ratio of peak stress in reduced ligament to the peak stress in normal ligament.

M total number of stress ranges at a point derived from the cycle counting procedure.

M_o longitudinal bending moment per unit length of circumference existing at the weld junction of layered spherical shells or heads due to discontinuity or external loads.

M_1 externally applied bending moment.

m material constant used for the fatigue knock-down factor used in the simplified elastic-plastic analysis

m_{ij} mechanical strain tensor, mechanical strain is defined as the total strain minus the free thermal strain.

m_{ss} exponent used in a fatigue analysis based on the structural stress.

n material constant used for the fatigue knock-down factor used in the simplified elastic-plastic analysis

n_k actual number of repetitions of the k^{th} cycle.

n_{css} material parameter for the cyclic stress-strain curve model.

N_k permissible number of cycles for the k^{th} cycle.

$N\left(C_1 S\right)$ number of cycles from the applicable design fatigue curve (see Annex 3.F, paragraph 3.F.1.2) evaluated at a stress amplitude of $C_1 S$.

$N\left(S_e\right)$ number of cycles from the applicable design fatigue curve (see Annex 3.F, paragraph 3.F.1.2) evaluated at a stress amplitude of S_e.

$N_{\Delta FP}$ design number of full-range pressure cycles including startup and shutdown.

$N_{\Delta P}$ number of significant cycles associated with ΔP_N.

$N_{\Delta PO}$ expected number of operating pressure cycles in which the range of pressure variation exceeds 20% of the design pressure for integral construction or 15% of the design pressure for non-integral construction.

$N_{\Delta S}$ number of significant cycles associated with ΔS_{ML}, significant cycles are those for which the range in temperature exceeds S_{as}.

$N_{\Delta T}$ number of cycles associated with ΔT_N.

$N_{\Delta TE}$ number of cycles associated with ΔT_E.

$N_{\Delta TM}$ number of significant cycles associated with ΔT_M.

$N_{\Delta TR}$ number of significant cycles associated with ΔT_R.

$N_{\Delta T\alpha}$ number of temperature cycles for components involving welds between materials having different coefficients of expansion.

ν Poisson's ratio.

P specified design pressure.

P_b primary bending equivalent stress.

P_m general primary membrane equivalent stress.

P_L local primary membrane equivalent stress.

Φ_B design factor for buckling.

Q	secondary equivalent stress resulting from operating loadings.
Q_1	externally applied shear force.
R	inside radius measured normal to the surface from the mid-wall of the shell to the axis of revolution , or the ratio of the minimum stress in the k^{th} cycle to the maximum stress in the k^{th} cycle, as applicable.
R_k	stress ratio for the k^{th} cycle.
$R_{b,k}$	ratio of the bending stress to the membrane plus bending stress.
$R_{b\tau,k}$	ratio of the bending component of the shear stress to the membrane plus bending component of the shear stress.
RSF	computed remaining strength factor.
R_1	mid-surface radius of curvature of region 1 where the local primary membrane stress exceeds $1.1S$.
R_2	mid-surface radius of curvature of region 2 where the local primary membrane stress exceeds $1.1S$.
S	allowable stress based on the material of construction and design temperature.
S_a	alternating stress obtained from a fatigue curve for the specified number of operating cycles.
S_{as}	stress amplitude from the applicable design fatigue curve (see Annex 3.F, paragraph 3.F.1.2) evaluated at 1E6 cycles.
S_e	computed equivalent stress.
S_Q	allowable limit on the secondary stress range.
S_{PS}	allowable limit on the primary plus secondary stress range (see paragraph 5.5.6.)
S_y	minimum specified yield strength at the design temperature.
S_y^L	specified plastic limit for limit-load analysis.
$S_{a,k}$	value of alternating stress obtained from the applicable design fatigue curve for the specified number of cycles of the k^{th} cycle.
$S_{alt,k}$	alternating equivalent stress for the k^{th} cycle.
$S_{y,k}$	yield strength of the material evaluated at the mean temperature of the k^{th} cycle.
$S_a(N)$	stress amplitude from the applicable design fatigue curve (see Annex 3.F, paragraph 3.F.1.2) evaluated at N cycles.
$S_a(N_{\Delta P})$	stress amplitude from the applicable design fatigue curve (see Annex 3.F, paragraph 3.F.1.2) evaluated at $N_{\Delta P}$ cycles.
$S_a(N_{\Delta S})$	stress amplitude from the applicable design fatigue curve (see Annex 3.F, paragraph 3.F.1.2) evaluated at $N_{\Delta S}$ cycles.
$S_a(N_{\Delta TN})$	stress amplitude from the applicable design fatigue curve (see Annex 3.F, paragraph 3.F.1.2) evaluated at $N_{\Delta TN}$ cycles.
$S_a(N_{\Delta TM})$	stress amplitude from the applicable design fatigue curve (see Annex 3.F, paragraph 3.F.1.2) evaluated at $N_{\Delta TM}$ cycles.

$S_a\left(N_{\Delta TR}\right)$ stress amplitude from the applicable design fatigue curve (see Annex 3.F, paragraph 3.F.1.2) evaluated at $N_{\Delta TR}$ cycles.

${}^m\sigma_{b,k}^e$ elastically calculated bending stress at the point under evaluation for the k^{th} cycle at the m point.

${}^n\sigma_{b,k}^e$ elastically calculated bending stress at the point under evaluation for the k^{th} cycle at the n point.

σ_e von Mises stress.

$\sigma_{e,k}$ von Mises stress for the k^{th} loading condition.

σ_i are the principal stress components.

$\sigma_{ij,k}$ stress tensor at the point under evaluation for the k^{th} cycle at the m point.

${}^m\sigma_{ij,k}$ stress tensor at the point under evaluation for the k^{th} cycle at the m point.

${}^n\sigma_{ij,k}$ stress tensor at the point under evaluation for the k^{th} cycle at the n point.

$\sigma_{ij,k}^{LT}$ stress tensor due to local thermal stress at the location and time point under evaluation for the k^{th} cycle.

${}^m\sigma_{m,k}^e$ elastically calculated membrane stress at the point under evaluation for the k^{th} cycle at the m point.

${}^n\sigma_{m,k}^e$ elastically calculated membrane stress at the point under evaluation for the k^{th} cycle at the n point.

$\sigma_{max,k}$ maximum stress in the k^{th} cycle.

$\sigma_{mean,k}$ mean stress in the k^{th} cycle.

$\sigma_{min,k}$ minimum stress in the k^{th} cycle.

σ_1 principal stress in the 1-direction.

σ_2 principal stress in the 2-direction.

σ_3 principal stress in the 3-direction.

$\sigma_{1,k}$ principal stress in the 1-direction for the k^{th} loading condition.

$\sigma_{2,k}$ principal stress in the 2-direction for the k^{th} loading condition.

$\sigma_{3,k}$ principal stress in the 3-direction for the k^{th} loading condition.

t minimum wall thickness in the region under consideration, or the thickness of the vessel, as applicable.

t_{ess} structural stress effective thickness.

t_1 minimum wall thickness associated with R_1.

t_2 minimum wall thickness associated with R_2.

${}^m\tau_{b,k}^e$ elastically calculated bending component of shear stress distribution at the point under evaluation for the k^{th} cycle at the m point.

${}^n\tau_{b,k}^e$ elastically calculated bending component of shear stress distribution at the point under evaluation for the k^{th} cycle at the n point.

${}^{m}\tau^{e}_{m,k}$	elastically calculated membrane component of shear stress distribution at the point under evaluation for the k^{th} cycle at the m point.
${}^{n}\tau^{e}_{m,k}$	elastically calculated membrane component of shear stress distribution at the point under evaluation for the k^{th} cycle at the n point.
UTS	minimum specified ultimate tensile strength at room temperature.
w	required with of attachment.
X	maximum general primary membrane stress divided by the yield strength.
YS	minimum specified yield strength at room temperature.

5.14 Tables

Table 5.1 – Loads And Load Cases To Be Considered In A Design

Loading Condition	Design Loads
Pressure Testing	1. Dead load of component plus insulation, fireproofing, installed internals, platforms and other equipment supported from the component in the installed position. 2. Piping loads including pressure thrust 3. Applicable live loads excluding vibration and maintenance live loads. 4. Pressure and fluid loads (water) for testing and flushing equipment and piping unless a pneumatic test is specified. 5. Wind loads
Normal Operation	1. Dead load of component plus insulation, refractory, fireproofing, installed internals, catalyst, packing, platforms and other equipment supported from the component in the installed position. 2. Piping loads including pressure thrust 3. Applicable live loads. 4. Pressure and fluid loading during normal operation. 5. Thermal loads.
Normal Operation plus Occasional (note: occasional loads are usually governed by wind and earthquake; however, other load types such as snow and ice loads may govern, see ASCE-7)	1. Dead load of component plus insulation, refractory, fireproofing, installed internals, catalyst, packing, platforms and other equipment supported from the component in the installed position. 2. Piping loads including pressure thrust 3. Applicable live loads. 4. Pressure and fluid loading during normal operation. 5. Thermal loads. 6. Wind, earthquake or other occasional loads, whichever is greater. 7. Loads due to wave action
Abnormal or Start-up Operation plus Occasional (see note above)	1. Dead load of component plus insulation, refractory, fireproofing, installed internals, catalyst, packing, platforms and other equipment supported from the component in the installed position. 2. Piping loads including pressure thrust 3. Applicable live loads. 4. Pressure and fluid loading associated with the abnormal or start-up conditions. 5. Thermal loads. 6. Wind loads.

Table 5.2 – Load Descriptions

Design Load Parameter	Description
P	Internal and external specified design pressure
P_s	Static head from liquid or bulk materials (e.g. catalyst)
D	Dead weight of the vessel, contents, and appurtenances at the location of interest, including the following: • Weight of vessel including internals, supports (e.g. skirts, lugs, saddles, and legs), and appurtenances (e.g. platforms, ladders, etc.) • Weight of vessel contents under operating and test conditions • Refractory linings, insulation • Static reactions from the weight of attached equipment, such as motors, machinery, other vessels, and piping
L	• Appurtenance Live loading • Effects of fluid momentum, steady state and transient
E	Earthquake loads (see ASCE 7 for the specific definition of the earthquake load, as applicable)
W	Wind Loads
W_{pt}	Is the pressure test wind load case. The design wind speed for this case shall be specified by the Owner-User.
S_s	Snow Loads
T	Is the self-restraining load case (i.e. thermal loads, applied displacements). This load case does not typically affect the collapse load, but should be considered in cases where elastic follow-up causes stresses that do not relax sufficiently to redistribute the load without excessive deformation.

Table 5.3 – Load Case Combinations and Allowable Stresses for an Elastic Analysis

Design Load Combination (1)	Allowable Stress
1) $P+P_s+D$	Determined based on the Stress Category shown in Figure 5.1
2) $P+P_s+D+L$	
3) $P+P_s+D+L+T$	
4) $P+P_s+D+S_s$	
5) $0.6D+(W\ or\ 0.7E)$ (2)	
6) $0.9P+P_s+D+(W\ or\ 0.7E)$	
7) $0.9P+P_s+D+0.75(L+T)+0.75S_s$	
8) $0.9P+P_s+D+0.75(W\ or\ 0.7E)+0.75L+0.75S_s$	

Notes

1) The parameters used in the Design Load Combination column are defined in Table 5.2.
2) This load combination addresses an overturning condition. If anchorage is included in the design, consideration of this load combination is not required
3) Loads listed herein shall be considered to act in the combinations described above; whichever produces the most unfavorable effect in the component being considered. Effects of one or more loads not acting shall be considered.

Table 5.4 – Load Case Combinations and Load Factors for a Limit Load Analysis

Design Conditions	
Criteria	Required Factored Load Combinations
Global Criteria	1) $1.5\left(P+P_s+D\right)$ 2) $1.3\left(P+P_s+D+T\right)+1.7L+0.54S_s$ 3) $1.3\left(P+P_s+D\right)+1.7S_s+\left(1.1L \text{ or } 0.86W\right)$ 4) $1.3\left(P+P_s+D\right)+1.7W+1.1L+0.54S_s$ 5) $1.3\left(P+P_s+D\right)+1.1E+1.1L+0.21S_s$
Local Criteria	Per Table 5.5
Serviceability Criteria	Per User's Design Specification, if applicable, see Table 5.5
Hydrostatic Test Conditions	
Global Criteria	$\max\left[1.43,\ 1.25\left(\dfrac{S_T}{S}\right)\right]\cdot\left(P+P_s+D\right)+2.6W_{pt}$
Serviceability Criteria	Per User's Design Specification, if applicable.
Pneumatic Test Conditions	
Global Criteria	$1.15\left(\dfrac{S_T}{S}\right)\cdot\left(P+P_s+D\right)+2.6W_{pt}$
Serviceability Criteria	Per User's Design Specification, if applicable.

Notes:

1) The parameters used in the Design Load Combination column are defined in Table 5.2.
2) See paragraph 5.2.3.4 for descriptions of global and serviceability criteria.
3) S is the allowable membrane stress at the design temperature.
4) S_T is the allowable membrane stress at the pressure test temperature.
5) Loads listed herein shall be considered to act in the combinations described above; whichever produces the most unfavorable effect in the component being considered. Effects of one or more loads not acting shall be considered.

Table 5.5 – Load Case Combinations and Load Factors for an Elastic-Plastic Analysis

Design Conditions	
Criteria	Required Factored Load Combinations
Global Criteria	1) $2.4(P + P_s + D)$ 2) $2.1(P + P_s + D + T) + 2.7L + 0.86S_s$ 3) $2.1(P + P_s + D) + 2.7S_s + (1.7L \text{ or } 1.4W)$ 4) $2.1(P + P_s + D) + 2.7W + 1.7L + 0.86S_s$ 5) $2.1(P + P_s + D) + 1.7E + 1.7L + 0.34S_s$
Local Criteria	$1.7(P + P_s + D)$
Serviceability Criteria	Per User's Design Specification, if applicable, see paragraph 5.2.4.3.b.
Hydrostatic Test Conditions	
Global and Local Criteria	$\max\left[2.3,\ 2.0\left(\frac{S_T}{S}\right)\right]\cdot(P + P_s + D) + W_{pt}$
Serviceability Criteria	Per User's Design Specification, if applicable.
Pneumatic Test Conditions	
Global and Local Criteria	$1.8\left(\frac{S_T}{S}\right)\cdot(P + P_s + D) + W_{pt}$
Serviceability Criteria	Per User's Design Specification, if applicable.

Notes:

1) The parameters used in the Design Load Combination column are defined in Table 5.2.
2) See paragraph 5.2.4.3 for descriptions of global and serviceability criteria.
3) S is the allowable membrane stress at the design temperature.
4) S_T is the allowable membrane stress at the pressure test temperature.
5) Loads listed herein shall be considered to act in the combinations described above; whichever produces the most unfavorable effect in the component being considered. Effects of one or more loads not acting shall be considered.

Table 5.6 – Examples Of Stress Classification

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
Any shell including cylinders, cones, spheres and formed heads	Shell plate remote from discontinuities	Internal pressure	General membrane Gradient through plate thickness	P_m Q
		Axial thermal gradient	Membrane Bending	Q Q
	Near nozzle or other opening	Net-section axial force and/or bending moment applied to the nozzle, and/or internal pressure	Local membrane Bending Peak (fillet or corner)	P_L Q F
	Any location	Temperature difference between shell and head	Membrane Bending	Q Q
	Shell distortions such as out-of-roundness and dents	Internal pressure	Membrane Bending	P_m Q
Cylindrical or conical shell	Any section across entire vessel	Net-section axial force, bending moment applied to the cylinder or cone, and/or internal pressure	Membrane stress averaged through the thickness, remote from discontinuities; stress component perpendicular to cross section	P_m
			Bending stress through the thickness; stress component perpendicular to cross section	P_b
	Junction with head or flange	Internal pressure	Membrane Bending	P_L Q
Dished head or conical head	Crown	Internal pressure	Membrane Bending	P_m P_b
	Knuckle or junction to shell	Internal pressure	Membrane Bending	P_L [note (1)] Q
Flat head	Center region	Internal pressure	Membrane Bending	P_m P_b
	Junction to shell	Internal pressure	Membrane Bending	P_L Q [note (2)]

Table 5.6 – Examples Of Stress Classification

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
Perforated head or shell	Typical ligament in a uniform pattern	Pressure	Membrane (averaged through cross section) Bending (averaged through width of ligament., but gradient through plate) Peak	P_m P_b F
	Isolated or atypical ligament	Pressure	Membrane Bending Peak	Q F F
Nozzle (see paragraph 5.6)	Within the limits of reinforcement given by paragraph 4.5	Pressure and external loads and moments including those attributable to restrained free end displacements of attached piping	General membrane Bending (other than gross structural discontinuity stresses) averaged through nozzle thickness	P_m P_m
	Outside the limits of reinforcement given by paragraph 4.5	Pressure and external axial, shear, and torsional loads including those attributable to restrained free end displacements of attached piping	General Membrane	P_m
		Pressure and external loads and moments, excluding those attributable to restrained free end displacements of attached piping	Membrane Bending	P_L P_b
		Pressure and all external loads and moments	Membrane Bending Peak	P_L Q F
	Nozzle wall	Gross structural discontinuities	Membrane Bending Peak	P_L Q F
		Differential expansion	Membrane Bending Peak	Q Q F
Cladding	Any	Differential expansion	Membrane Bending	F F
Any	Any	Radial temperature distribution [note (3)]	Equivalent linear stress [note (4)]	Q

Table 5.6 – Examples Of Stress Classification

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
			Nonlinear portion of stress distribution	F
Any	Any	Any	Stress concentration (notch effect)	F

Notes:

1. Consideration shall be given to the possibility of wrinkling and excessive deformation in vessels with large diameter-to-thickness ratio.
2. If the bending moment at the edge is required to maintain the bending stress in the center region within acceptable limits, the edge bending is classified as P_b; otherwise, it is classified as Q.
3. Consider possibility of thermal stress ratchet.
4. Equivalent linear stress is defined as the linear stress distribution that has the same net bending moment as the actual stress distribution.

Table 5.7 – Uniaxial Strain Limit for use in Multiaxial Strain Limit Criterion

Material	Maximum Temperature	ε_{Lu} Uniaxial Strain Limit (1), (2), (3)			α_{sl}
		m_2	Elongation Specified	Reduction of Area Specified	
Ferritic Steel	480°C (900°F)	$0.60\left(1.00-R\right)$	$2\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2
Stainless Steel and Nickel Base Alloys	480°C (900°F)	$0.75\left(1.00-R\right)$	$3\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	0.6
Duplex Stainless Steel	480°C (900°F)	$0.70\left(0.95-R\right)$	$2\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2
Super Alloys (4)	480°C (900°F)	$1.90\left(0.93-R\right)$	$\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2
Aluminum	120°C (250°F)	$0.52\left(0.98-R\right)$	$1.3\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2
Copper	65°C (150°F)	$0.50\left(1.00-R\right)$	$2\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2
Titanium and Zirconium	260°C (500°F)	$0.50\left(0.98-R\right)$	$1.3\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2

Notes:

1. If the elongation and reduction in area are not specified, then $\varepsilon_{Lu}=m_2$. If the elongation or reduction in area is specified, then ε_{Lu} is the maximum number computed from columns 3, 4 or 5, as applicable.
2. R is the ratio of the minimum specified yield strength divided by the minimum specified ultimate tensile strength.
3. E is the % elongation and RA is the % reduction in area determined from the applicable material specification.
4. Precipitation hardening austenitic alloys

Table 5.8 – Temperature Factors For Fatigue Screening Criteria

Metal temperature Differential		Temperature Factor For Fatigue Screening Criteria
°C	°F	
28 or less	50 or less	0
29 to 56	51 to 100	1
57 to 83	101 to 150	2
84 to 139	151 to 250	4
140 to 194	251 to 350	8
195 to 250	351 to 450	12
Greater than 250	Greater than 450	20

Notes:

1. If the weld metal temperature differential is unknown or cannot be established, a value of 20 shall be used.
2. As an example illustrating the use of this table, consider a component subject to metal temperature differentials for the following number of thermal cycles.

Temperature Differential	Temperature Factor Based On Temperature Differential	Number Of Thermal Cycles
28 °C (50 °F)	0	1000
50 °C (90 °F)	1	250
222 °C (400 °F)	12	5

The effective number of thermal cycles due to changes in metal temperature is:

$$N_{\Delta TE} = 1000(0) + 250(1) + 5(12) = 310 \ cycles$$

Table 5.9 – Fatigue Screening Criteria For Method A

Description		
Integral Construction	Attachments and nozzles in the knuckle region of formed heads	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \le 350$
	All other components	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \le 1000$
Non-integral construction	Attachments and nozzles in the knuckle region of formed heads	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \le 60$
	All other components	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \le 400$

Table 5.10 – Fatigue Screening Criteria Factors For Method B

Description		C_1	C_2
Integral Construction	Attachments and nozzles in the knuckle region of formed heads	4	2.7
	All other components	3	2
Non-integral construction	Attachments and nozzles in the knuckle region of formed heads	5.3	3.6
	All other components	4	2.7

Table 5.11 – Weld Surface Fatigue-Strength-Reduction Factors

Weld Condition	Surface Condition	Quality Levels (see Table 5.12)						
		1	2	3	4	5	6	7
Full penetration	Machined	1.0	1.5	1.5	2.0	2.5	3.0	4.0
	As-welded	1.2	1.6	1.7	2.0	2.5	3.0	4.0
Partial Penetration	Final Surface Machined	NA	1.5	1.5	2.0	2.5	3.0	4.0
	Final Surface As-welded	NA	1.6	1.7	2.0	2.5	3.0	4.0
	Root	NA	NA	NA	NA	NA	NA	4.0
Fillet	Toe machined	NA	NA	1.5	NA	2.5	3.0	4.0
	Toe as-welded	NA	NA	1.7	NA	2.5	3.0	4.0
	Root	NA	NA	NA	NA	NA	NA	4.0

Table 5.12 – Weld Surface Fatigue-Strength-Reduction Factors

Fatigue-Strength-Reduction Factor	Quality Level	Definition
1.0	1	Machined or ground weld that receives a full volumetric examination, and a surface that receives MT/PT examination and a VT examination.
1.2	1	As-welded weld that receives a full volumetric examination, and a surface that receives MT/PT and VT examination
1.5	2	Machined or ground weld that receives a partial volumetric examination, and a surface that receives MT/PT examination and VT examination
1.6	2	As-welded weld that receives a partial volumetric examination, and a surface that receives MT/PT and VT examination
1.5	3	Machined or ground weld surface that receives MT/PT examination and a VT examination (visual), but the weld receives no volumetric examination inspection
1.7	3	As-welded or ground weld surface that receives MT/PT examination and a VT examination (visual), but the weld receives no volumetric examination inspection
2.0	4	Weld has received a partial or full volumetric examination, and the surface has received VT examination, but no MT/PT examination
2.5	5	VT examination only of the surface; no volumetric examination nor MT/PT examination.
3.0	6	Volumetric examination only
4.0	7	Weld backsides that are non-definable and/or receive no examination.

Notes:

1. Volumetric examination is RT or UT in accordance with Part 7.
2. MT/PT examination is magnetic particle or liquid penetrant examination in accordance with Part 7
3. VT examination is visual examination in accordance with Part 7.
4. See WRC Bulletin 432 for further information.

Table 5.13 – Fatigue Penalty Factors For Fatigue Analysis

Material	K_e (1)		T_{max} (2)	
	m	n	(^oC)	(^oF)
Low alloy steel	2.0	0.2	371	700
Martensitic stainless steel	2.0	0.2	371	700
Carbon steel	3.0	0.2	371	700
Austenitic stainless steel	1.7	0.3	427	800
Nickel-chromium-iron	1.7	0.3	427	800
Nickel-copper	1.7	0.3	427	800

Notes:

1. Fatigue penalty factor
2. The fatigue penalty factor should only be used if all of the following are satisfied:
   - The component is not subject to thermal ratcheting, and
   - The maximum temperature in the cycle is within the value in the table for the material.

5.15 Figures

Stress Category	Primary			Secondary Membrane plus Bending	Peak
	General Membrane	Local Membrane	Bending		
Description (For examples, see Table 5.2)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Average stress across any solid section. Considers discontinuities but not concentrations. Produced only by mechanical loads.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical load or by differential thermal expansion. Excludes local stress concentrations.	1. Increment added to primary or secondary stress by a concentration (notch). 2. Certain thermal stresses which may cause fatigue but not distortion of vessel shape.
Symbol	P_m	P_L	P_b	Q	F

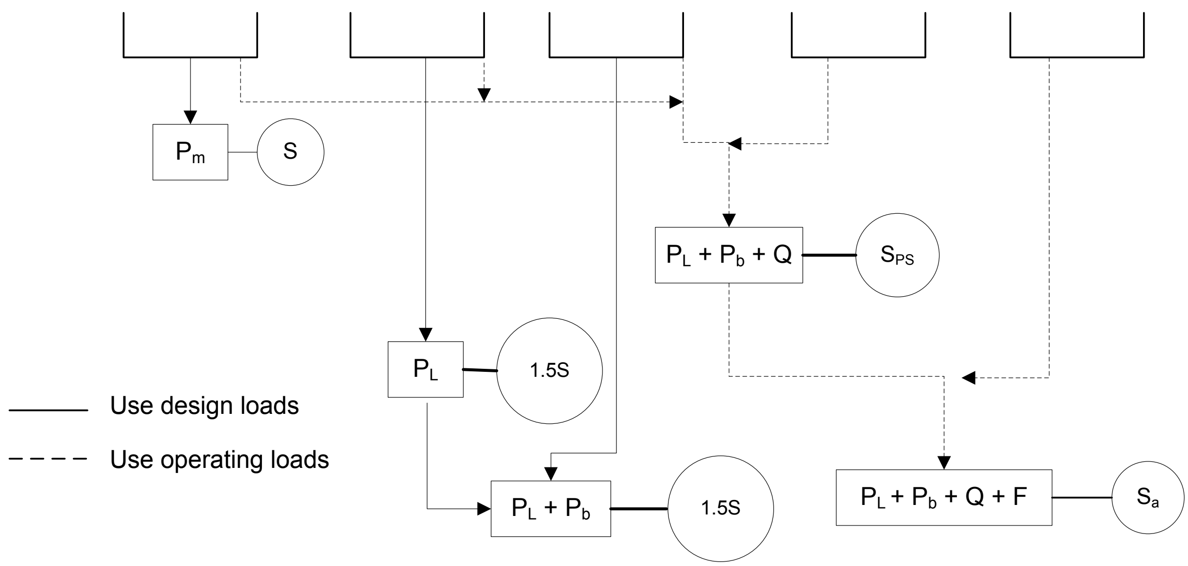


Figure 5.1
Stress Categories and Limits of Equivalent Stress

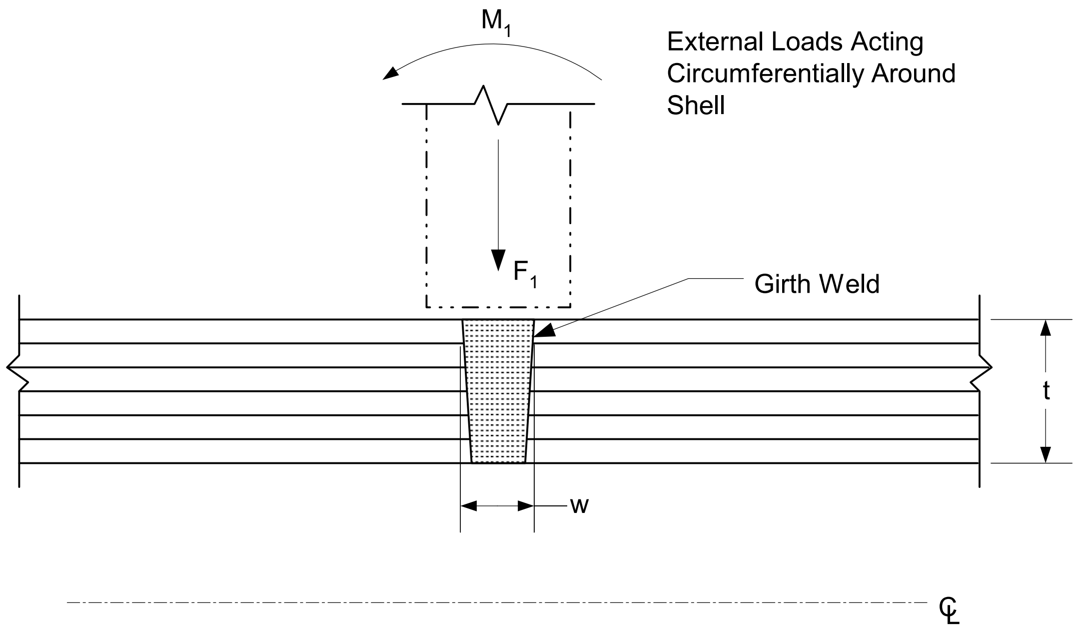


Figure 5.2
Example of Girth Weld Used to Tie Layers for Solid Wall Equivalence

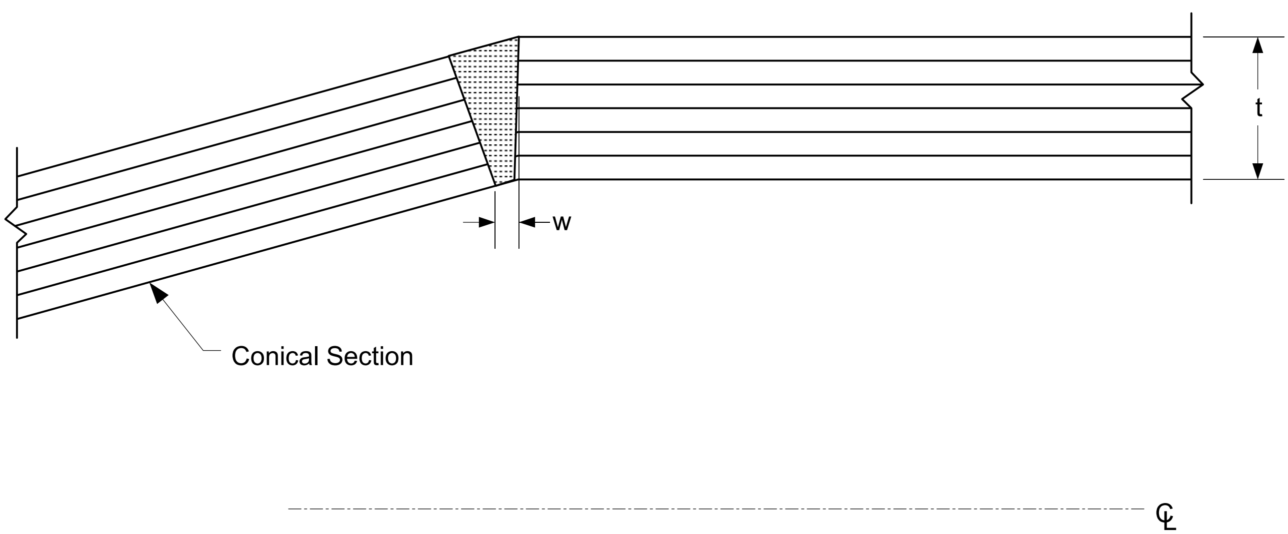


Figure 5.3
Example of Circumferential Butt Weld Attachment Between Layered Sections in Zone of Discontinuity

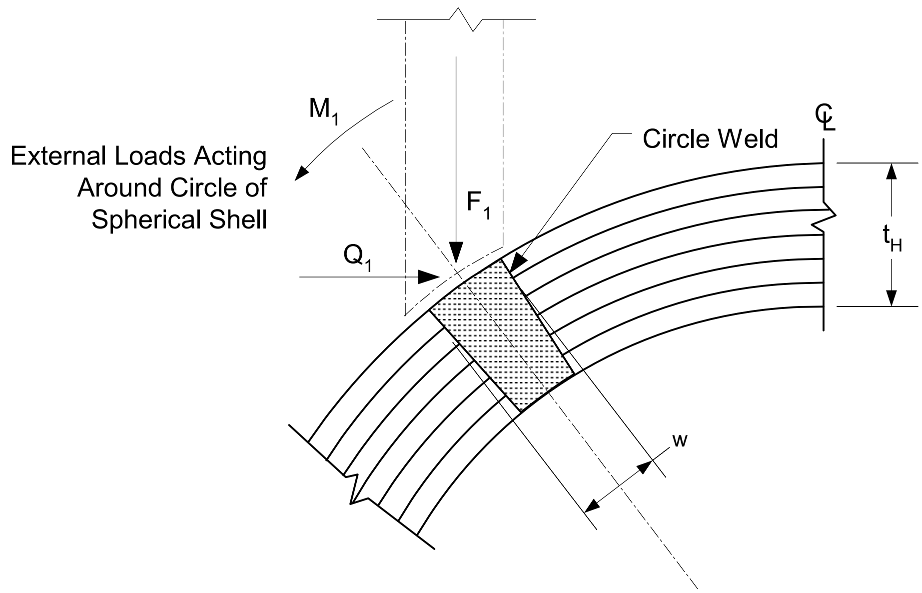


Figure 5.4
An Example of Circle Weld Used to Tie Layers for Solid Wall Equivalence

ANNEX 5.A

LINEARIZATION OF STRESS RESULTS FOR STRESS CLASSIFICATION

(INFORMATIVE)

5.A.1 Scope

This Annex provides recommendations for post-processing of the results from an elastic finite element stress analysis for comparison to the limits in paragraph 5.2.2.

5.A.2 General

a) In the finite element method, when continuum elements are used in an analysis, the total stress distribution is obtained. Therefore, to produce membrane and bending stresses, the total stress distribution shall be linearized on a stress component basis and used to calculate the equivalent stresses. If shell elements (shell theory) are used, then the membrane and bending stresses shall be obtained directly from shell stress resultants.

b) Membrane and bending stresses are developed on cross sections through the thickness of a component. These sections are called stress classification planes (SCPs). In a planar geometry, a Stress Classification Line (SCL) is obtained by reducing two opposite sides of a SCP to an infinitesimal length. SCPs are flat planes that cut through a section of a component and SCLs are straight lines that cut through a section of a component. SCLs are surfaces when viewed in an axisymmetric or planar geometry. Examples of an SCP and SCL are given in Figure 5.A.1 and Figure 5.A.2.

c) The following three approaches are provided for linearization of finite element results.

   1) Stress Integration Method – This method can be used to linearize stress results from continuum finite element models [Ref. WRC-429].

   2) Structural Stress Method Based on Nodal Forces – This method is based on processing of nodal forces, and has been shown to be mesh insensitive and correlate well with welded fatigue data [Ref. WRC-474].

   3) Structural Stress Method Based on Stress Integration – This method utilizes the Stress Integration Method, but restricts the set of elements that contribute to the line of nodes being processed.

d) The Structural Stress Method based on Stress Integration is recommended unless another method can be shown to produce a more accurate assessment for the given component and loading condition. This method matches the Structural Stress Method Based on Nodal Forces, which is insensitive to mesh refinement. In addition, this method can be performed with post-processing tools typically provided by commercial finite element analysis software.

5.A.3 Selection of Stress Classification Lines

a) Pressure vessels usually contain structural discontinuity regions where abrupt changes in geometry, material or loading occur. These regions are typically the locations of highest stress in a component. For the evaluation of failure modes of plastic collapse and ratcheting, Stress Classification Lines (SCLs) are typically located at gross structural discontinuities. For the evaluation of local failure and fatigue, SCLs are typically located at local structural discontinuities.

b) For SCLs that span a material discontinuity (e.g. base metal with cladding), the SCL should include all materials and associated loadings. If one of the materials, such as cladding, is neglected for strength calculations, then only the base metal thickness should be used to calculate the membrane and bending stresses from the linearized forces and moments across the full section for the evaluation of plastic collapse.

c) To most accurately determine the linearized membrane and bending stresses for comparison to elastic stress limits, the following guidelines should be followed. These guidelines can be used as a qualitative means to evaluate the applicability of different SCLs. Failure to comply with any of these criteria may not produce valid membrane and/or bending stresses. Application of the limit load or elastic-plastic analysis methods in Part 5 is recommended for cases where elastic stress analysis and stress linearization may produce ambiguous results.

   1) SCLs should be oriented normal to contour lines of the stress component of highest magnitude. However, as this may be difficult to implement, similar accuracy can be obtained by orienting the SCL normal to the mid-surface of the cross section. SCL orientation guidelines are shown in Figure 5.A.3.

   2) Hoop and meridional component stress distributions on the SCL should be monotonically increasing or decreasing, except for the effects of stress concentration or thermal peak stresses, see Figure 5.A.3.b.

   3) The distribution of the through-thickness stress should be monotonically increasing or decreasing. For pressure loading the through-thickness stress should be equal to the compressive pressure on the applied surface, and approximately zero on the other surface defining the SCL (see Figure 5.A.3.c). When the SCL is not perpendicular to the surfaces, this requirement will not be satisfied.

   4) The shear stress distribution should be parabolic and/or the stress should be low relative to the hoop and meridional stresses. Depending on the type of loading, the shear stress should be approximately zero on both surfaces defined by the SCL. Guidelines are provided in Figure 5.A.3.d.

      i) The shear stress distribution along an SCL will approximate a parabolic distribution only when the inner and outer surfaces are parallel and the SCL is normal to the surfaces. If the surfaces are not parallel or an SCL is not normal to the surfaces, the appropriate shear distribution will not be obtained. However, if the magnitude of shear stress is small as compared to the hoop or meridional stresses, this orientation criterion can be waived.

      ii) When the shear stress distribution is approximately linear, the shear stress is likely to be significant.

   5) For pressure boundary components, the hoop or meridional stresses typically are the largest magnitude component stresses and are the dominant terms in the equivalent stress. Typically the hoop or meridional stresses deviate from a monotonically increasing or decreasing trend along an SCL if the SCL is skewed with respect to the interior, exterior, or mid surfaces. For most pressure vessel applications, the hoop or meridional stresses due to pressure should be nearly linear.

5.A.4 Stress Integration Method

5.A.4.1 Continuum Elements

5.A.4.1.1 Overview

Stress results derived from a finite element analysis utilizing two-dimensional or three-dimensional continuum elements may be processed using the stress integration method. Stress components are integrated along SCLs through the wall thickness to determine the membrane and bending stress components. The peak stress components can be derived directly using this procedure by subtracting the membrane plus bending stress distribution from the total stress distribution. Using these components, the equivalent stress shall be computed per Equation (5.1).

5.A.4.1.2 Stress Linearization Procedure

The methods to derive the membrane, bending, and peak components of a stress distribution are shown below, and in Figure 5.A.4. The component stresses used for the calculations shall be based on a local coordinate system defined by the orientation of the SCL, see Figure 5.A.2.

a) STEP 1 – Calculate the membrane stress tensor. The membrane stress tensor is the tensor comprised of the average of each stress component along the stress classification line, or:

$$\sigma_{ij,m} = \frac{1}{t}\int_0^t \sigma_{ij}\,dx \tag{5.A.1}$$

b) STEP 2 – Calculate the bending stress tensor.

1) Bending stresses are calculated only for the local hoop and meridional (normal) component stresses, and not for the local component stress parallel to the SCL or in-plane shear stress.

2) The linear portion of shear stress needs to be considered only for shear stress distributions that result in torsion of the SCL (out-of-plane shear stress in the normal-hoop plane, see Figure 5.A.2).

3) The bending stress tensor is comprised of the linear varying portion of each stress component along the stress classification line, or:

$$\sigma_{ij,b} = \frac{6}{t^2}\int_0^t \sigma_{ij}\left(\frac{t}{2}-x\right)dx \tag{5.A.2}$$

c) STEP 3 – Calculate the peak stress tensor. The peak stress tensor is the tensor whose components are equal to:

$$\sigma_{ij,F}\left(x\right)\Big|_{x=0} = \sigma_{ij}\left(x\right)\Big|_{x=0} - \left(\sigma_{ij,m}+\sigma_{ij,b}\right) \tag{5.A.3}$$

$$\sigma_{ij,F}\left(x\right)\Big|_{x=t} = \sigma_{ij}\left(x\right)\Big|_{x=t} - \left(\sigma_{ij,m}-\sigma_{ij,b}\right) \tag{5.A.4}$$

d) Step 4 – Calculate the three principal stresses at the ends of the SCL based on components of membrane and membrane plus bending stresses.

e) Step 5 – Calculate the equivalent stresses using Equation (5.1) at the ends of the SCL based on components of membrane and membrane plus bending stresses.

5.A.4.2 Shell Elements

5.A.4.2.1 Overview

Stress results derived from a finite element analysis utilizing two-dimensional or three-dimensional shells are obtained directly from the analysis results. Using the component stresses, the equivalent stress shall be computed per Equation (5.1).

5.A.4.2.2 Stress Linearization Procedure

The methods to derive the membrane, bending, and peak components of a stress distribution are shown below.

a) The membrane stress tensor is the tensor comprised of the average of each stress component along the stress classification line, or:

$$\sigma_{ij,m} = \frac{\sigma_{ij,in} + \sigma_{ij,out}}{2} \tag{5.A.5}$$

b) The bending stress tensor is the tensor comprised of the linear varying portion of each stress component along the stress classification line, or:

$$\sigma_{ij,b} = \frac{\sigma_{ij,in} - \sigma_{ij,out}}{2} \tag{5.A.6}$$

c) The peak stress tensor is the tensor whose components are equal to:

$$\sigma_{ij,F} = \left(\sigma_{ij,m} + \sigma_{ij,b}\right)\left(K_f - 1\right) \tag{5.A.7}$$

5.A.5 Structural Stress Method Based on Nodal Forces

5.A.5.1 Overview

Stress results derived from a finite element analysis utilizing continuum or shell elements may be processed using the Structural Stress Method based on nodal forces. The mesh-insensitive structural stress method provides a robust procedure for capturing the membrane and bending stresses and can be directly utilized in fatigue design of welded joints. With this method, the structural stress normal to a hypothetical cracked plane at a weld is evaluated. For typical pressure vessel component welds, the choice of possible crack orientations is straightforward (e.g. toe of fillet weld). Two alternative calculation procedures for the structural stress method are presented for continuum elements; a procedure based on nodal forces and a procedure based on stress integration. A typical finite element continuum model and stress evaluation line for this type of analysis is shown in Figure 5.A.5.

5.A.5.2 Continuum Elements

a) Stress results derived from a finite element analysis utilizing two-dimensional or three-dimensional continuum elements may be processed using the structural stress method and nodal forces as described below. The membrane and bending stresses can be computed from element nodal internal forces using the equations provided in Table 5.A.1. The process is illustrated in Figure 5.A.6. This method is recommended when internal force results can be obtained as part of the finite element output because the results are insensitive to the mesh density.

b) When using three-dimensional continuum elements, forces and moments must be summed with respect to the mid-thickness of a member from the forces at nodes in the solid model at a through-thickness cross section of interest. For a second order element, three summation lines of nodes are processed along the element faces through the wall thickness. The process is illustrated in Figure 5.A.7.

c) For a symmetric structural stress range, the two weld toes have equal opportunity to develop fatigue cracks. Therefore, the structural stress calculation involves establishing the equilibrium equivalent membrane and bending stress components with respect to one-half of the plate thickness. The equivalent structural stress calculation procedure for a symmetric stress state is illustrated in Figure 5.A.8.

5.A.5.3 Shell Elements

a) Stress results derived from a finite element analysis utilizing shell elements may be processed using the structural stress method and nodal forces. The membrane and bending stresses can be computed from element nodal internal forces using the equations provided in Table 5.A.2. A typical shell model is illustrated in Figure 5.A.9.

b) When using three-dimensional shell elements, forces and moments with respect to the mid-thickness of a member must be obtained at a cross section of interest. The process is illustrated in Figure 5.A.10.

5.A.6 Structural Stress Method Based on Stress Integration

As an alternative to the nodal force method above, stress results derived from a finite element analysis utilizing two-dimensional or three-dimensional continuum elements may be processed using the Structural Stress Method Based on Stress Integration. This method utilizes the Stress Integration Method of paragraph 5.A.3, but restricts the set of elements that contribute to the line of nodes being processed. The elements applicable to the SCL for the region being evaluated shall be included in the post-processing, as is illustrated in Figure 5.A.11.

5.A.7 Nomenclature

σ_s	structural stress
$\Delta\sigma_s$	structural stress range.
f_i	line force at element location position i.
NF_j	nodal force at node j, normal to the section.
NF_{ij}	nodal force at node j, normal to the section, for element location position i.
NM_j	in-plane nodal moment at node j, normal to the section, for a shell element.
F_i	nodal force resultant for element location position i.
K_f	fatigue strength reduction factor used to compute the cyclic stress amplitude or range.
m_i	line moment at element location position i.
n	number of nodes in the through-wall thickness direction.
M_i	nodal moment resultant for element location position i.
σ_m	membrane stress.
σ_b	bending stress.
σ_{ij}	stress tensor at the point under evaluation.
$\sigma_{ij,m}$	membrane stress tensor at the point under evaluation.
$\sigma_{ij,b}$	bending stress tensor at the point under evaluation.
$\sigma_{ij,F}$	peak stress component.
$\sigma_{ij,in}$	stress tensor on the inside surface of the shell.
$\sigma_{ij,out}$	stress tensor on the outside surface of the shell.
σ_{mi}	membrane stress for element location position i.
σ_{bi}	bending stress for element location position i.
r_j	radial coordinate of node j for an axisymmetric element.

s_j	local coordinate, parallel to the stress classification line, that defines the location of nodal force NF_j relative to the mid-thickness of the section.
P	primary equivalent stress.
Q	secondary equivalent stress.
X_L	local X axis, oriented parallel to the stress classification line.
Y_L	local Y axis, oriented normal to the stress classification line.
X_g	global X axis.
Y_g	global Y axis.
t	minimum wall thickness in the region under consideration, or the thickness of the vessel, as applicable.
w	width of the element to determine structural stresses from Finite Element Analysis.
x	through-wall thickness coordinate.

5.A.8 Tables

Table 5.A.1 – Structural Stress Definitions For Continuum Finite Elements

Element Type	Membrane Stress	Bending Stress
Two-Dimensional Axisymmetric Second Order (8-Node) Continuum Elements	$\sigma_m = \frac{1}{t}\sum\frac{NF_j}{2\pi r_j}$	$\sigma_b = \frac{6}{t^2}\sum\frac{NF_j \cdot s_j}{2\pi r_j}$
Two-Dimensional Second Order Plane Stress or Plane Strain (8-Node) Continuum Elements	$\sigma_m = \frac{1}{t}\sum\frac{NF_j}{w}$	$\sigma_b = \frac{6}{t^2}\sum\frac{NF_j \cdot s_j}{w}$
Three-Dimensional Second Order (20-Node) Continuum Elements	$\sigma_{mi} = \frac{f_i}{t}$ Note: f_i represents the line force corresponding to the element location positions ($i = 1,2,3$) along the element width (w); position $i = 2$ corresponds to the mid-side of the element (see Figure 5.A.7): $f_1 = \frac{3(6F_1 + 2F_3 - F_2)}{2w}$ $f_2 = \frac{-3(2F_1 + 2F_3 - 3F_2)}{4w}$ $f_3 = \frac{3(2F_1 + 6F_3 - F_2)}{2w}$ In the above, F_1, F_2, and F_3 are the nodal force resultants (producing normal membrane stress to Section A-A) through the thickness and along the width (w) of the group of elements $F_i = \sum NF_{ij}$ summed over the nodes from $j = 1, n$ (number of nodes in the through-thickness direction) at Section A-A (see Figure 5.A.7)	$\sigma_{bi} = \frac{6 \cdot m_i}{t^2}$ Note: m_i represents the line moment corresponding to the element location positions ($i = 1,2,3$) along the element width (w); position $i = 2$ corresponds to the mid-side of the element (see Figure 5.A.7): $m_1 = \frac{3(6M_1 + 2M_3 - M_2)}{2w}$ $m_2 = \frac{-3(2M_1 + 2M_3 - 3M_2)}{4w}$ $m_3 = \frac{3(2M_1 + 6M_3 - M_2)}{2w}$ In the above, M_1, M_2, and M_3 are the nodal moment resultants (producing normal bending stress to Section A-A) calculated based on nodal forces with respect to the mid-thickness (s_j) along the width (w) of the group of elements $M_i = \sum NF_{ij} \cdot s_j$ summed over the nodes from $j = 1, n$ (number of nodes in the through-thickness direction) at Section A-A (see Figure 5.A.7)

Table 5.A.2 – Structural Stress Definitions For Shell Or Plate Finite Elements

Element Type	Membrane Stress	Bending Stress
Three-Dimensional Second Order (8-Node) Shell Elements	$$\sigma_{mi} = \frac{f_i}{t}$$ Note: f_i represents the force corresponding to the element location positions ($i=1,2,3$) along the element width (w); position $i=2$ corresponds to the mid-side of the element (see Figure 5.A.10): $$f_1 = \frac{3(6NF_1 + 2NF_3 - NF_2)}{2w}$$ $$f_2 = \frac{-3(2NF_1 + 2NF_3 - 3NF_2)}{4w}$$ $$f_3 = \frac{3(2NF_1 + 6NF_3 - NF_2)}{2w}$$ In the above, NF_1, NF_2 , and NF_3 are the internal nodal forces (in the direction normal to Section A-A) from the shell model along a weld (see Figure 5.A.10)	$$\sigma_{bi} = \frac{6 \cdot m_i}{t^2}$$ Note: m_i represents the moment corresponding to the element location positions ($i=1,2,3$) along the element width (w); position $i=2$ corresponds to the mid-side of the element (see Figure 5.A.10): $$m_1 = \frac{3(6NM_1 + 2NM_3 - NM_2)}{2w}$$ $$m_2 = \frac{-3(2NM_1 + 2NM_3 - 3NM_2)}{4w}$$ $$m_3 = \frac{3(2NM_1 + 6NM_3 - NM_2)}{2w}$$ In the above, NM_1, NM_2 , and NM_3 are the internal nodal moments (producing normal bending stresses to Section A-A) from the shell model along a weld (see Figure 5.A.10)
Three-Dimensional First Order (4-Node) Shell Elements	$$\sigma_{mi} = \frac{f_i}{t}$$ Note: f_i represents the force corresponding to the element corner node location positions ($i=1,2$) along the element width (w): $$f_1 = \frac{2(2NF_1 - NF_2)}{w}$$ $$f_2 = \frac{2(2NF_2 - NF_1)}{w}$$	$$\sigma_{bi} = \frac{6 \cdot m_i}{t^2}$$ Note: m_i represents the moment corresponding to the element corner node location positions ($i=1,2$) along the element width (w): $$m_1 = \frac{2(2NM_1 - NM_2)}{w}$$ $$m_2 = \frac{2(2NM_2 - NM_1)}{w}$$
Axisymmetric Linear and Parabolic Shell Finite Element	$$\sigma_m = \frac{NF_j}{2\pi r_j t}$$	$$\sigma_b = \frac{6 \cdot NM_j}{2\pi r_j t^2}$$

5.A.9 Figures

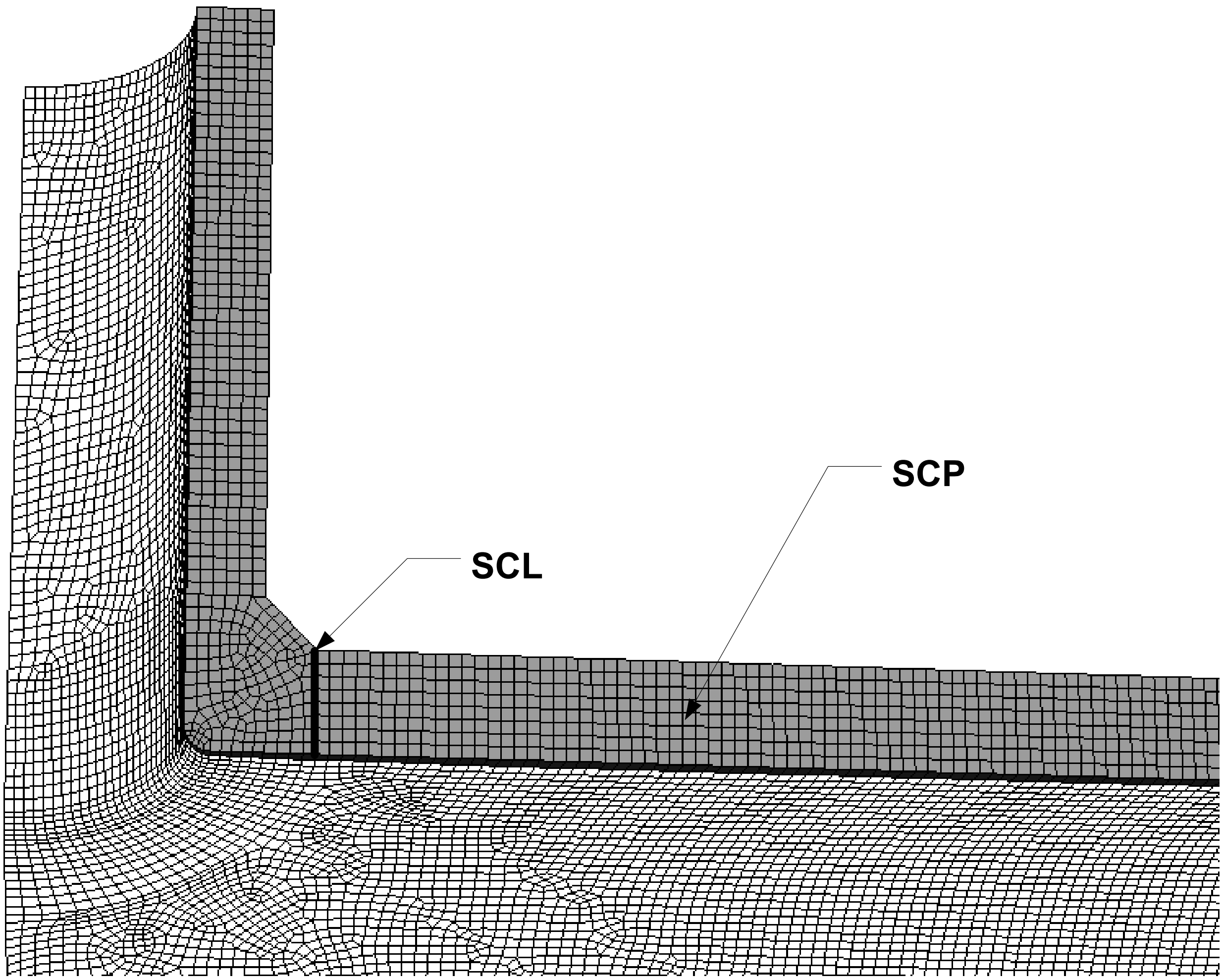
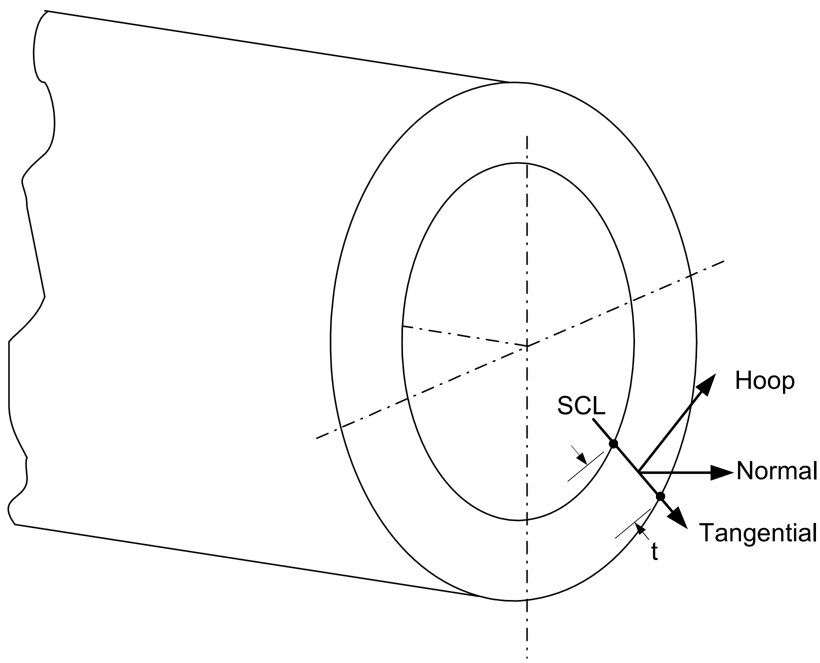
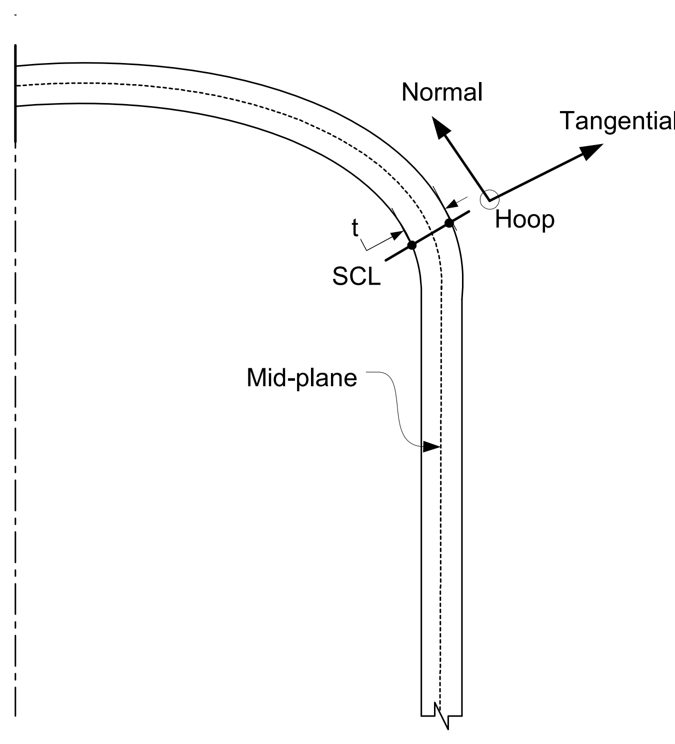


Figure 5.A.1
Stress Classification Line (SCL) and Stress Classification Plane (SCP)



(a) SCL Orientation, Three-Dimensional Model



(b) SCL Orientation, Two-Dimensional Model

Figure 5.A.2
Stress Classification Lines (SCLs)

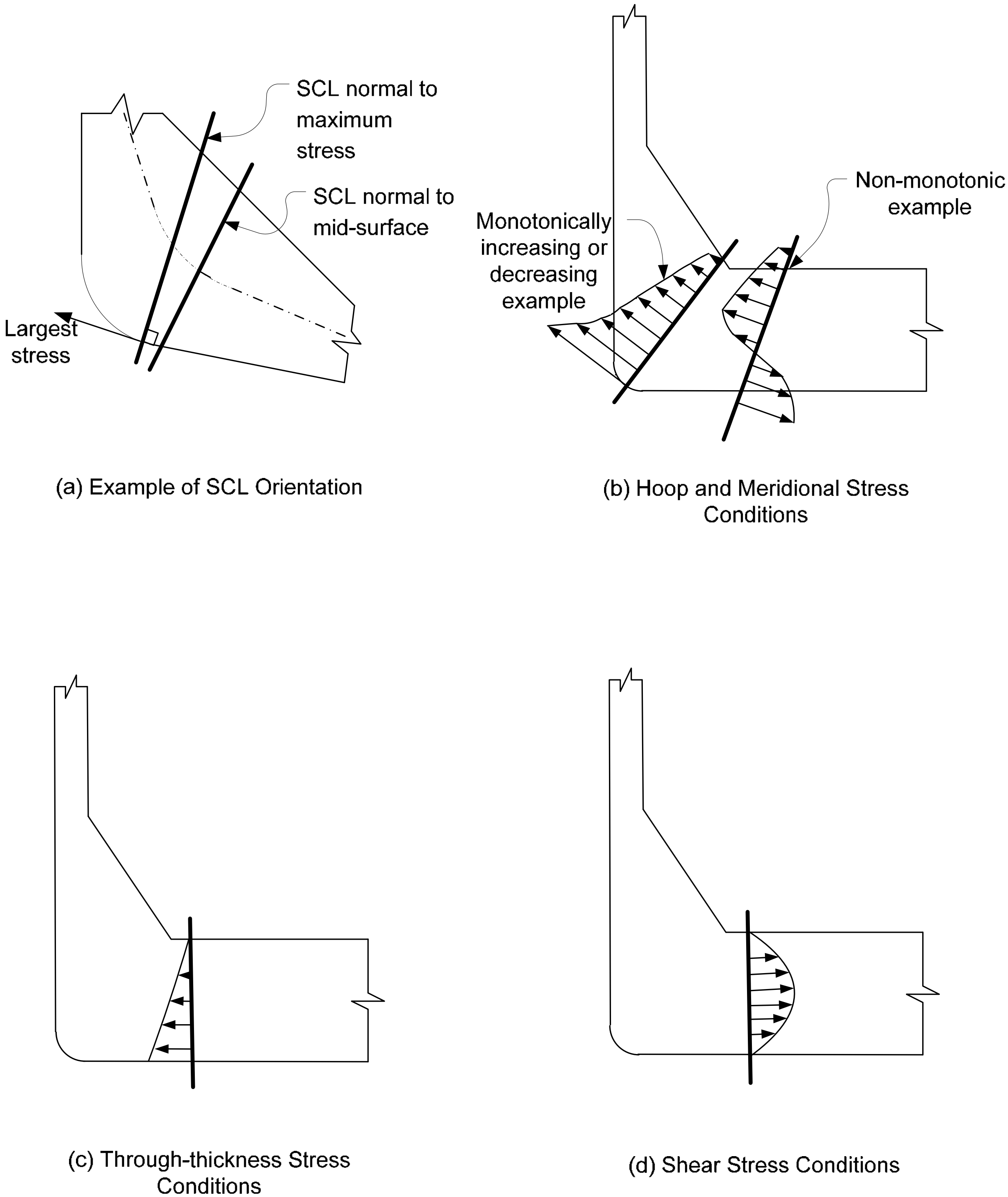


Figure 5.A.3
Stress Classification Line Orientation and Validity Guidelines

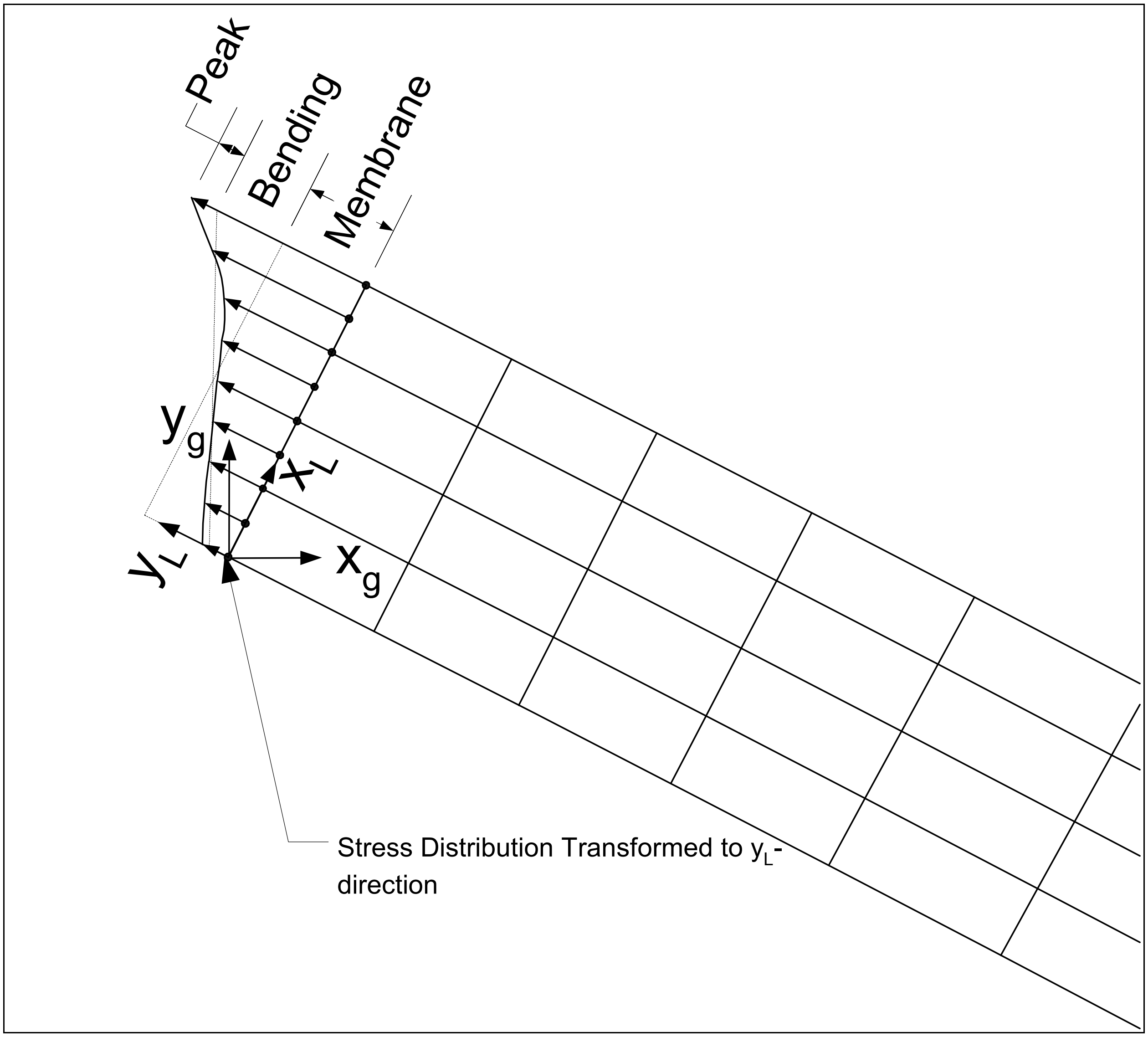
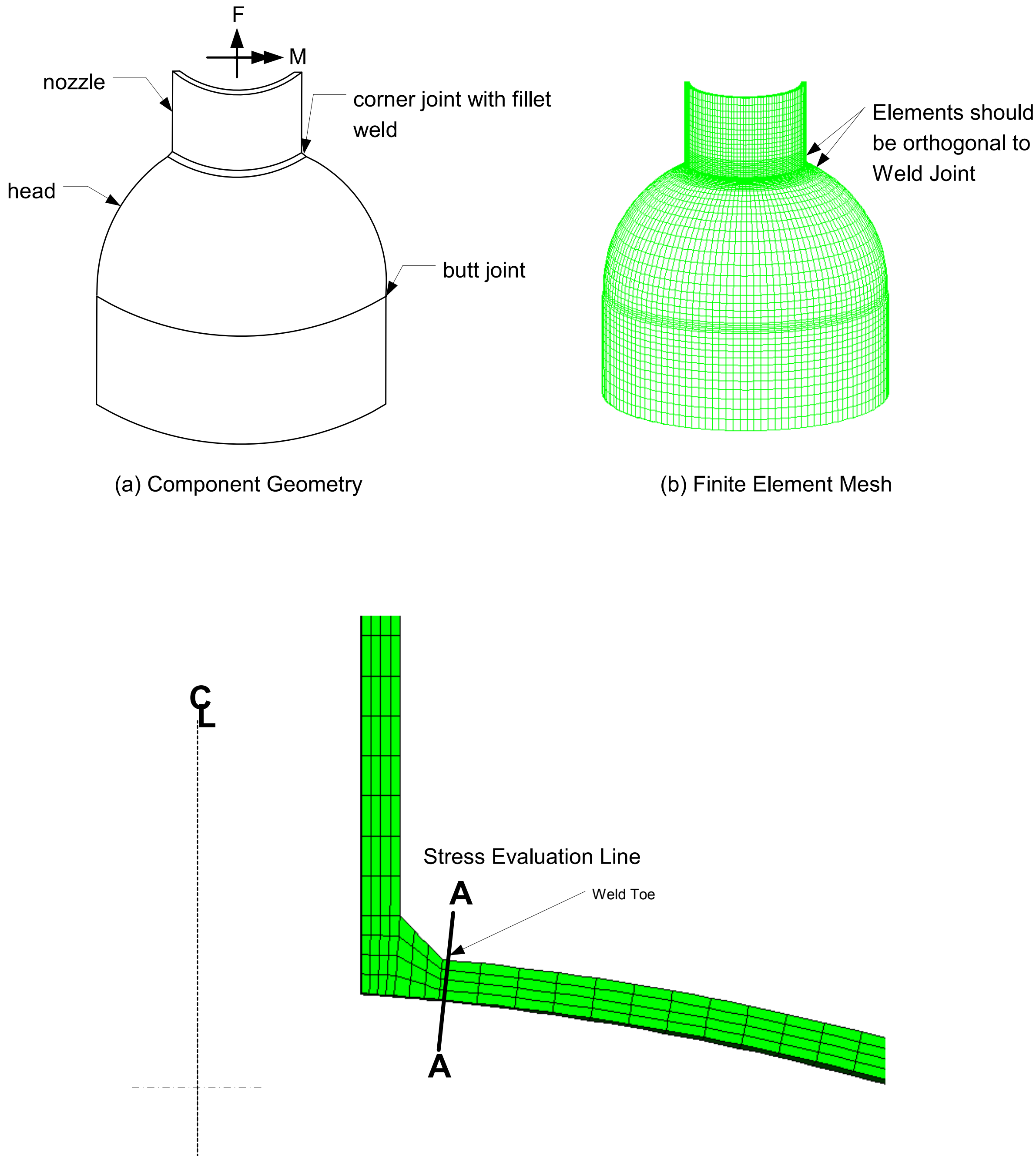


Figure 5.A.4
Computation of Membrane and Bending Equivalent Stresses by the Stress Integration Method Using the Results from a Finite Element Model with Continuum Elements



(a) Component Geometry

(b) Finite Element Mesh

(c) Section A-A Stress Classification LIne

Figure 5.A.5
Continuum Finite Element Model Stress Classification Line for the Structural Stress Method

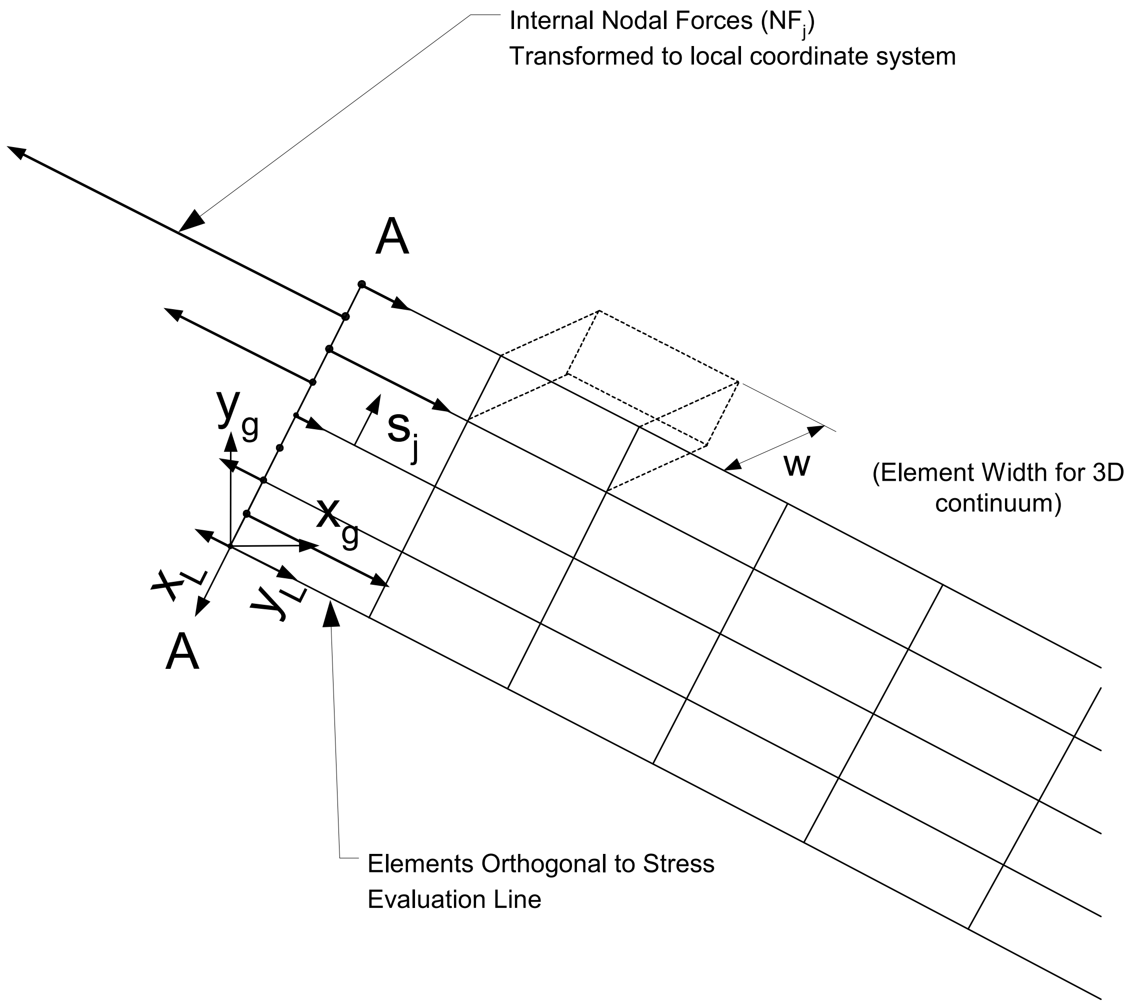


Figure 5.A.6
Computation of Membrane and Bending Equivalent Stresses by the Structural Stress Method Using Nodal Force Results from a Finite Element Model with Continuum Elements

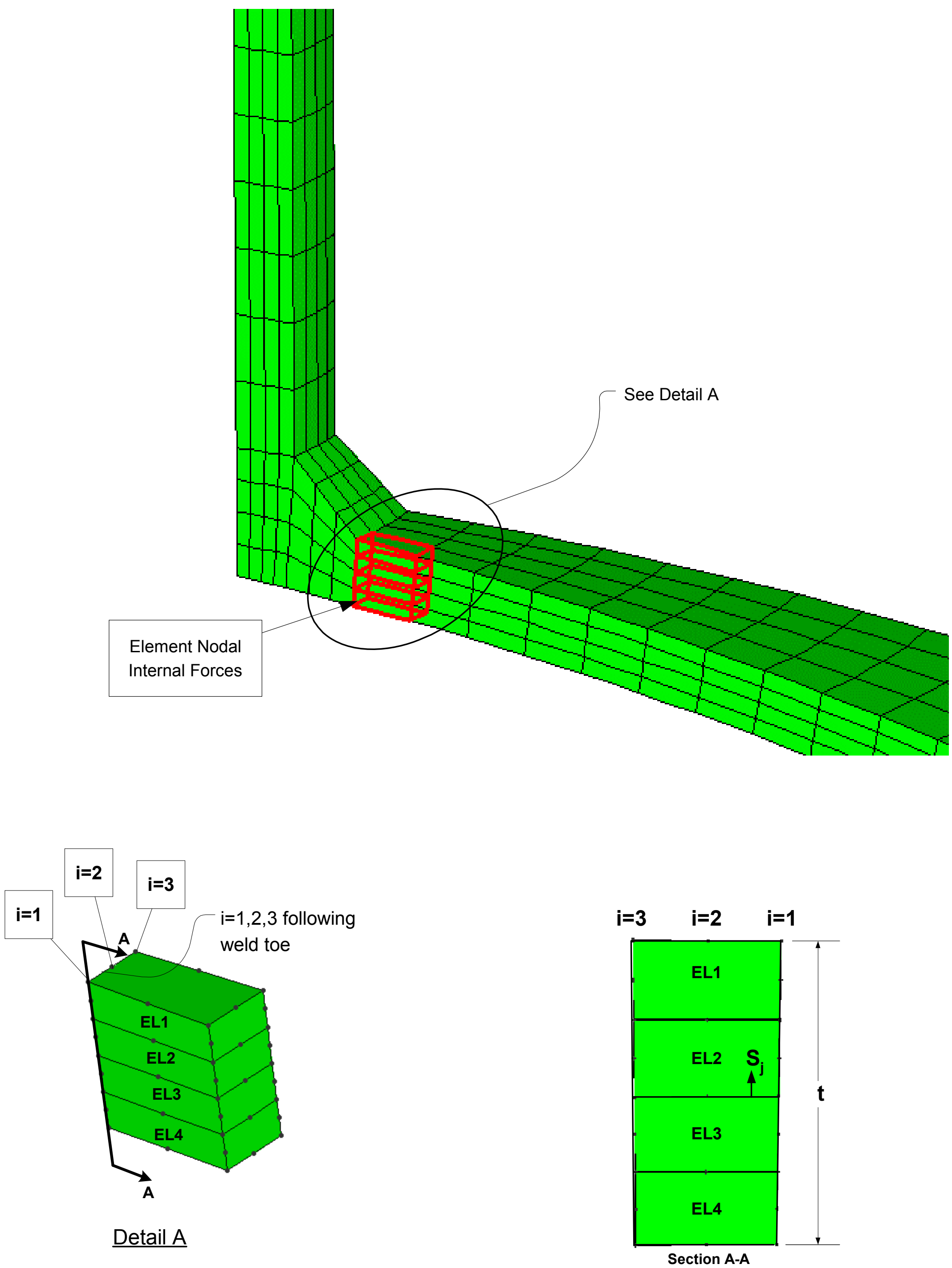
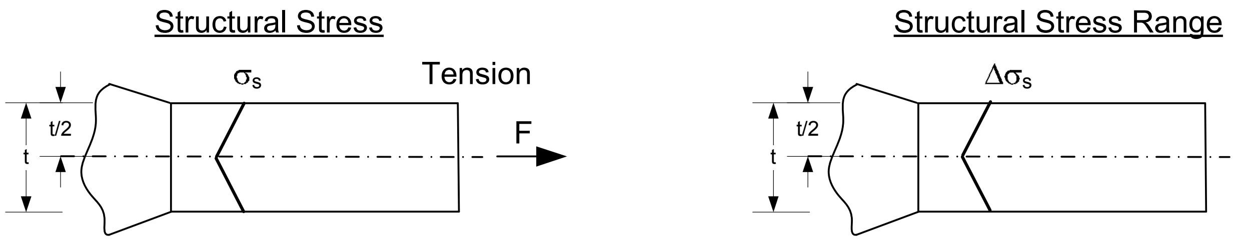
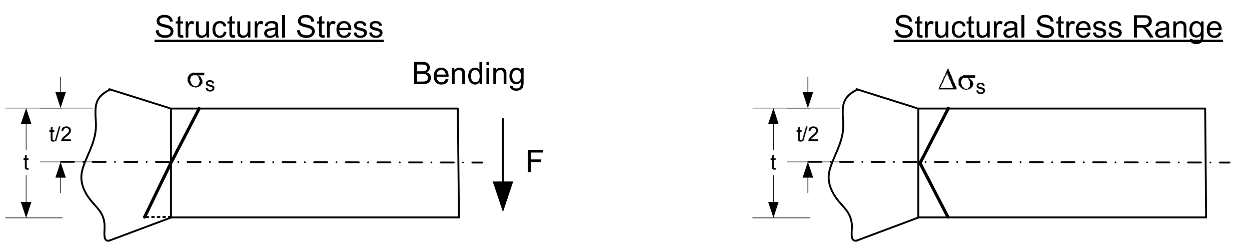


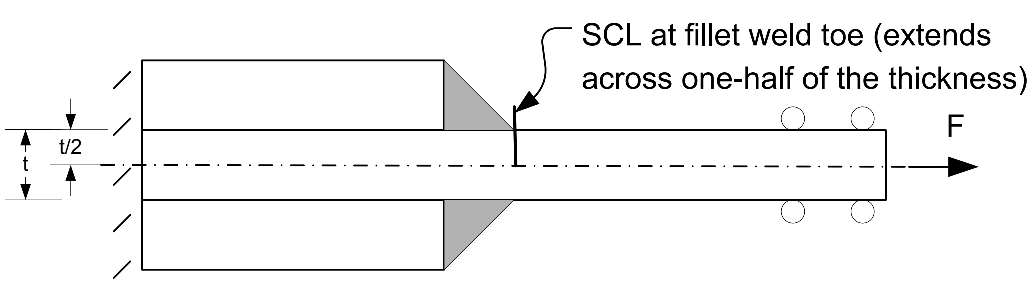
Figure 5.A.7
Processing Nodal Force Results with the Structural Stress Method Using the Results from a Finite Element Model with Three Dimensional Second Order Continuum Elements



(a) Symmetric Structural Stress State (symmetric joint and symmetric loading)

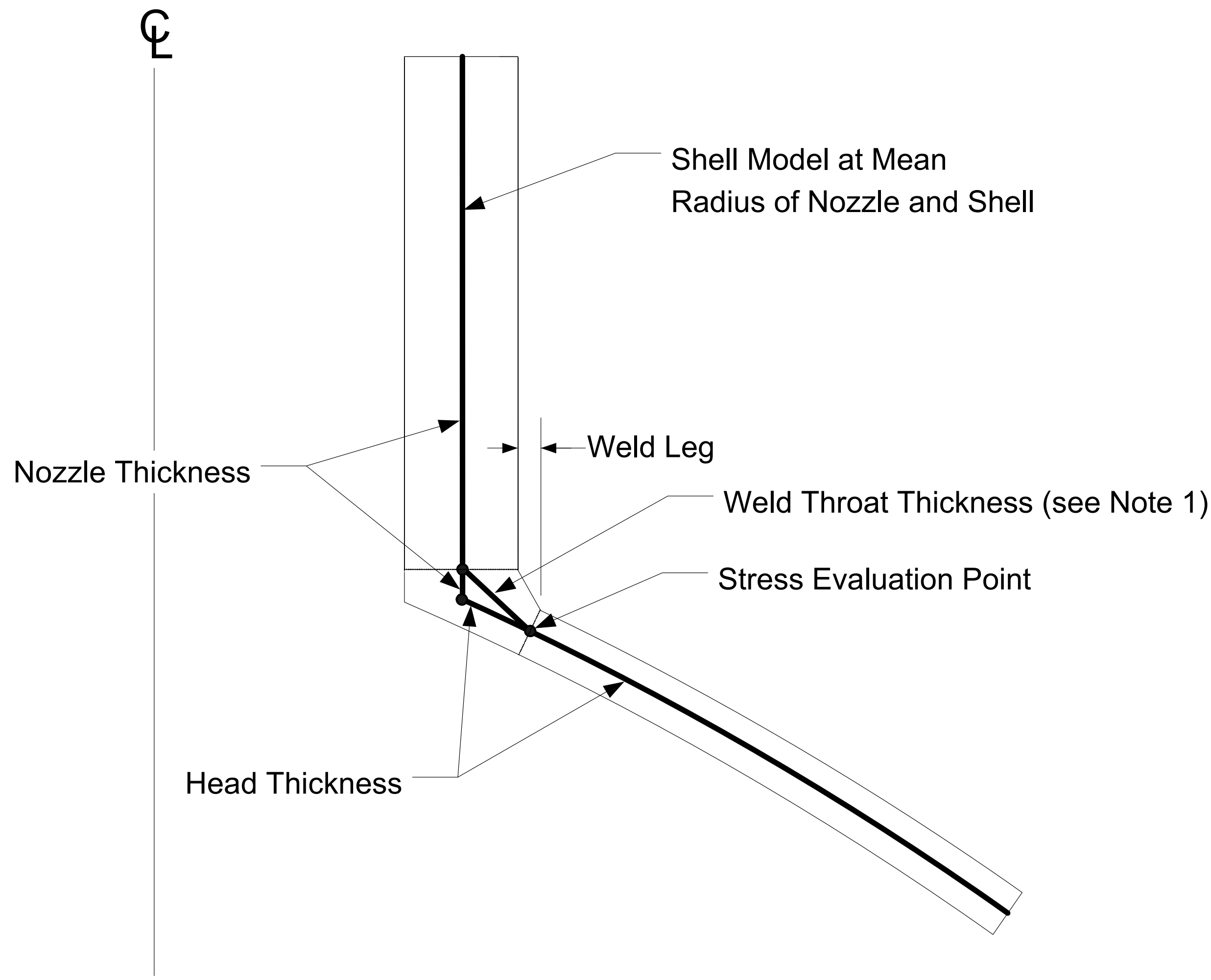


(b) Anti-symmetric Structural Stress State (symmetric joint and anti-symmetric loading)



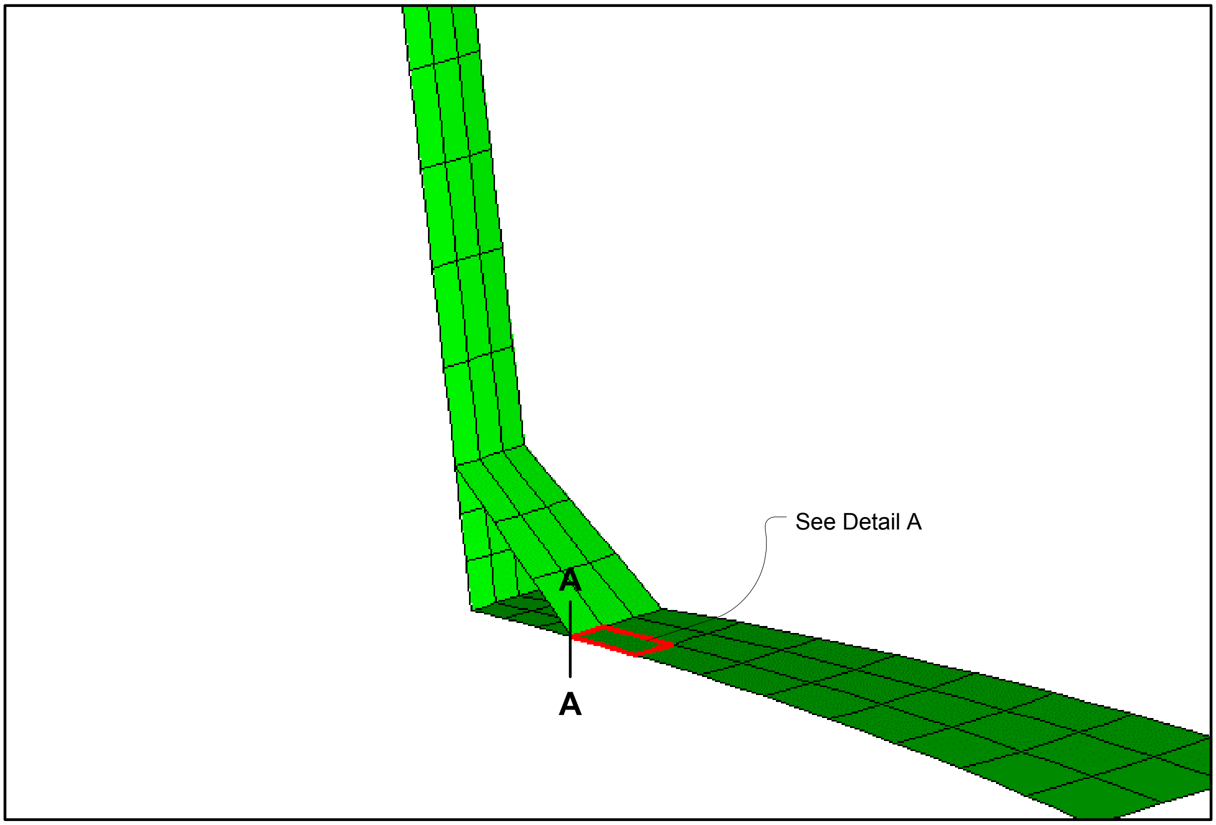
(c) Example of Symmetric Joint (double plate lap fillet weld)

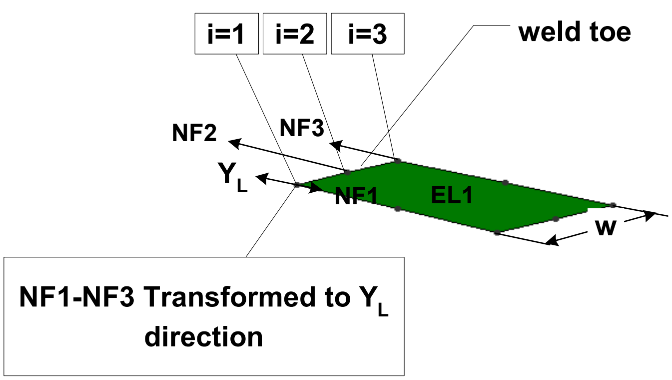
Figure 5.A.8
Processing Structural Stress Method Results for a Symmetric Structural Stress Range



Note 1: The thickness and material properties of the shell element used to model the fillet weld should be established based on producing an equivalent stiffness of the actual fillet weld.

Figure 5.A.9
Computation of Membrane and Bending Equivalent Stresses by the Structural Stress Method Using the Results from a Finite Element Model with Shell Elements





Detail A

Figure 5.A.10
Processing Nodal Force Results with the Structural Stress Method Using the Results from a Finite Element Model With Three Dimensional Second Order Shell Elements

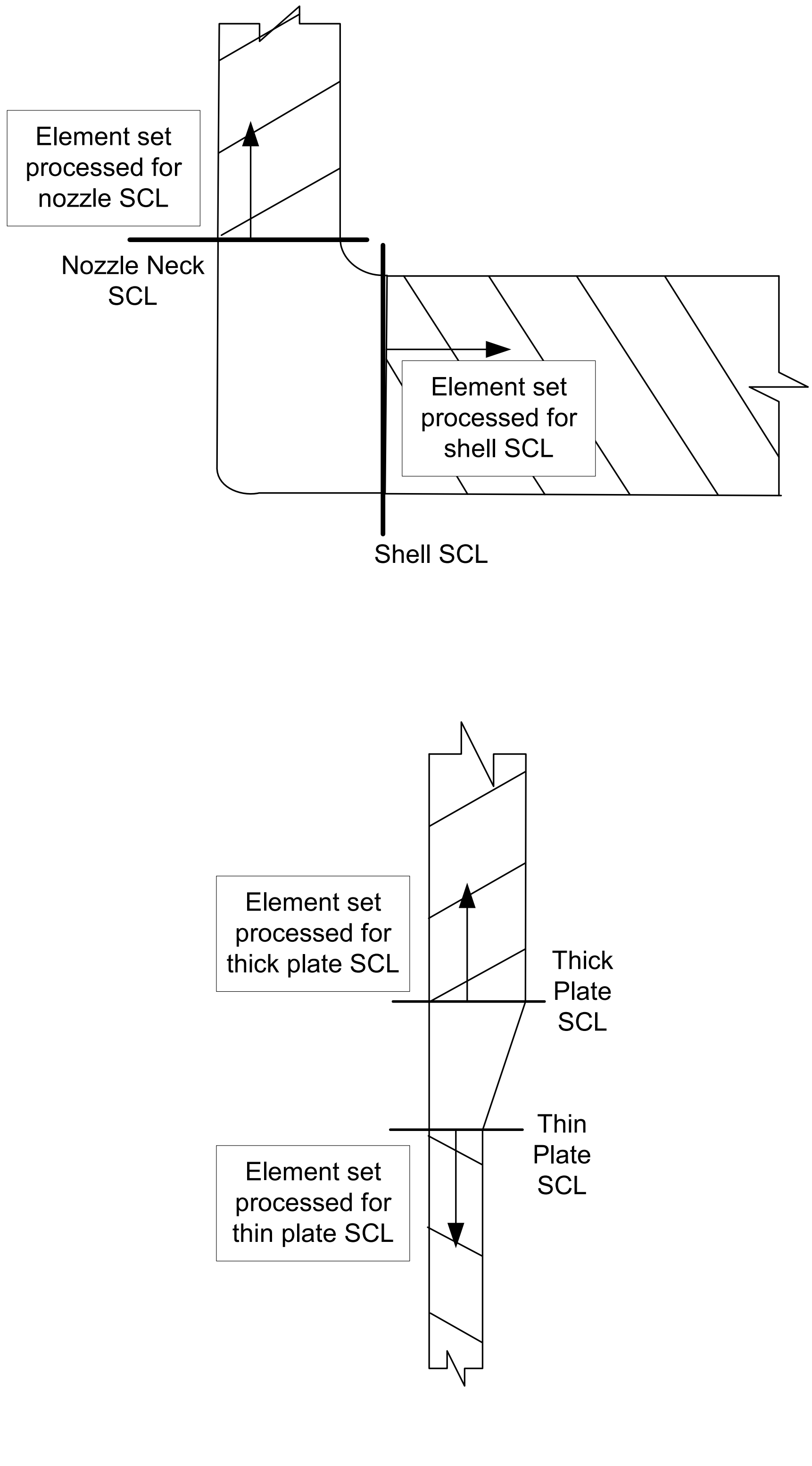


Figure 5.A.11
Element Sets for Processing Finite Element Nodal Stress Results with the Structural Stress Method Based on Stress Integration

ANNEX 5.B

HISTOGRAM DEVELOPMENT AND CYCLE COUNTING FOR FATIGUE ANALYSIS

(INFORMATIVE)

5.B.1 General

This annex contains cycle counting procedures required to perform a fatigue assessment for irregular stress or strain versus time histories. These procedures are used to break the loading history down into individual cycles that can be evaluated using the fatigue assessment rules of Part 5. Two cycle counting methods are presented in this Appendix. An alternative cycle counting method may be used if agreed to by the Owner-User.

5.B.2 Definitions

The definitions used in this Annex are shown below.

a) Event – The Users' Design Specification may include one or more events that produce fatigue damage. Each event consists of loading components specified at a number of time points over a time period and is repeated a specified number of times. For example, an event may be the startup, shutdown, upset condition, or any other cyclic action. The sequence of multiple events may be specified or random.

b) Cycle – A cycle is a relationship between stress and strain that is established by the specified loading at a location in a vessel or component. More than one stress-strain cycle may be produced at a location, either within an event or in transition between two events, and the accumulated fatigue damage of the stress-strain cycles determines the adequacy for the specified operation at that location. This determination shall be made with respect to the stabilized stress-strain cycle.

c) Proportional Loading – During constant amplitude loading, as the magnitudes of the applied stresses vary with time, the size of Mohr's circle of stress also varies with time. In some cases, even though the size of Mohr's circle varies during cyclic loading, if the orientation of the principal axes remains fixed, the loading is called proportional. An example of proportional loading is a shaft subjected to in-phase torsion and bending, where the ratio of axial and torsional stress remains constant during cycling.

d) Non-Proportional Loading – If the orientation of the principal axes are not fixed, but change orientation during cyclic loading, the loading is called non-proportional. An example of non-proportional loading is a shaft subjected to out-of-phase torsion and bending, where the ratio of axial and torsional stress varies continuously during cycling.

e) Peak – The point at which the first derivative of the loading or stress histogram changes from positive to negative.

f) Valley – The point at which the first derivative of the loading or stress histogram changes from negative to positive.

5.B.3 Histogram Development

5.B.3.1 The loading histogram should be determined based on the specified loadings provided in the Users' Design Specification. The loading histogram should include all significant operating loads and events that are applied to the component. The following should be considered in developing the loading histogram.

a) The number of repetitions of each event during the operation life.

b) The sequence of events during the operation life, if applicable.

c) Applicable loadings such as pressure, temperature, supplemental loads such as weight, support displacements, and nozzle reaction loadings.

d) The relationship between the applied loadings during the time history.

5.B.4 Cycle Counting Using the Rainflow Method

5.B.4.1 The Rainflow Cycle Counting Method (ASTM Standard No. E1049) is recommended to determine the time points representing individual cycles for the case of situations where the variation in time of loading, stress, or strain can be represented by a single parameter. This cycle counting method is not applicable for non-proportional loading. Cycles counted with the Rainflow Method correspond to closed stress-strain hysteresis loops, with each loop representing a cycle.

5.B.4.2 Recommended Procedure

a) STEP 1 – Determine the sequence of peaks and valleys in the loading histogram. If multiple loadings are applied, it may be necessary to determine the sequence of peaks and valleys using a stress histogram. If the sequence of events is unknown, the worst case sequence should be chosen.

b) STEP 2 – Re-order the loading histogram to start and end at either the highest peak or lowest valley, so that only full cycles are counted. Determine the sequence of peaks and valleys in the loading history. Let X denote the range under consideration, and let Y denote the previous range adjacent to X.

c) STEP 3 – Read the next peak or valley. If out of data, go to STEP 8.

d) STEP 4 – If there are less than 3 points, go to STEP 3; If not, form ranges X and Y using the three most recent peaks and valleys that have not been discarded.

e) STEP 5 – Compare the absolute values of ranges X and Y.

   1) If $X < Y$ go to STEP 3

   2) If $X \geq Y$ go to STEP 6

f) STEP 6 – Count range Y as one cycle; discard the peak and valley of Y. Record the time points and loadings or component stresses, as applicable, at the starting and ending time points of the cycle.

g) STEP 7 – Return to STEP 4 and repeat STEPs 4 to 6 until no more time points with stress reversals remain.

h) STEP 8 – Using the data recorded for the counted cycles perform fatigue assessment in accordance with Part 5.

5.B.5 Cycle Counting Using Max-Min Cycle Counting Method

5.B.5.1 Overview

The Max-Min Cycle Counting Method is recommended to determine the time points representing individual cycles for the case of non-proportional loading. The cycle counting is performed by first constructing the largest possible cycle, using the highest peak and lowest valley, followed by the second largest cycle, etc., until all peak counts are used.

5.B.5.2 Recommended Procedure

a) STEP 1 – Determine the sequence of peaks and valleys in the loading history. If some events are known to follow each other, group them together but otherwise arrange the random events in any order.

b) STEP 2 – Calculate the elastic stress components σ_{ij} produced by the applied loading at every point in time during each event at a selected location of a vessel. All stress components must be referred to the same global coordinate system. The stress analysis must include peak stresses at local discontinuities.

c) STEP 3 – Scan the interior points of each event and delete the time points at which none of the stress components indicate reversals (peaks or valleys).

d) STEP 4 – Using the stress histogram from STEP 2, determine the time point with the highest peak or lowest valley. Designate the time point as ${}^{m}t$, and the stress components as ${}^{m}\sigma_{ij}$.

e) STEP 5 – If time point ${}^{m}t$ is a peak in the stress histogram, determine the component stress range between time point ${}^{m}t$ and the next valley in the stress histogram. If time point ${}^{m}t$ is a valley, determine the component stress range between time point ${}^{m}t$ and the next peak. Designate the next time point as ${}^{n}t$, and the stress components as ${}^{n}\sigma_{ij}$. Calculate the stress component ranges and the von Mises equivalent stress range between time points ${}^{m}t$ and ${}^{n}t$.

$$ {}^{mn}\Delta\sigma_{ij} = {}^{m}\sigma_{ij} - {}^{n}\sigma_{ij} \tag{5.B.1} $$

$$ {}^{mn}\Delta S_{range} = \frac{1}{\sqrt{2}} \begin{bmatrix} \left({}^{mn}\Delta\sigma_{11} - {}^{mn}\Delta\sigma_{22}\right)^2 + \left({}^{mn}\Delta\sigma_{22} - {}^{mn}\Delta\sigma_{33}\right)^2 + \\ \left({}^{mn}\Delta\sigma_{33} - {}^{mn}\Delta\sigma_{11}\right)^2 + 6\left({}^{mn}\Delta\sigma_{12}^2 + {}^{mn}\Delta\sigma_{23}^2 + {}^{mn}\Delta\sigma_{31}^2\right) \end{bmatrix}^{0.5} \tag{5.B.2} $$

f) STEP 6 – Repeat STEP 5, for the current time point, ${}^{m}t$ and the time point of the next peak or valley in the sequence of the stress histogram. Repeat this process for every remaining time point in the stress histogram.

g) STEP 7 – Determine the maximum von Mises equivalent stress range obtained in STEP 5 and record the time points ${}^{m}t$ and ${}^{n}t$ that define the start and end points of the k^{th} cycle.

h) STEP 8 – Determine the event or events to which the time points ${}^{m}t$ and ${}^{n}t$ belong and record their specified number of repetitions as ${}^{m}N$ and ${}^{n}N$, respectively.

i) STEP 9 – Determine the number of repetitions of the k^{th} cycle.

1) If ${}^{m}N < {}^{n}N$: Delete the time point ${}^{m}t$ from those considered in STEP 4, and reduce the number of repetitions at time point ${}^{n}t$ from ${}^{n}N$ to $({}^{n}N - {}^{m}N)$.

2) If ${}^{m}N > {}^{n}N$: Delete the time point ${}^{n}t$ from those considered in STEP 4, and reduce the number of repetitions at time point ${}^{m}t$ from ${}^{m}N$ to $({}^{m}N - {}^{n}N)$.

3) If ${}^{m}N = {}^{n}N$: Delete both time points ${}^{m}t$ and ${}^{n}t$ from those considered in STEP 4.

j) STEP 10 – Return to STEP 4 and repeat STEPs 4 to 10 until no more time points with stress reversals remain.

k) STEP 11 – Using the data recorded for the counted cycles, perform fatigue assessment in accordance with Part 5. Note that an elastic-plastic fatigue assessment (see Part 5, paragraph 5.5.4) may be applied if ${}^{mn}\Delta S_{range}$ exceeds the yield point of the cyclic stress range-strain range curve of the material.

5.B.6 Nomenclature

$^{mn}\Delta S_{range}$ von Mises equivalent stress range between time points ^{m}t and ^{n}t.

σ_{ij} stress tensor at the point under evaluation.

$^{m}\sigma_{ij}$ stress tensor at the point under evaluation at time point ^{m}t.

$^{n}\sigma_{ij}$ stress tensor at the point under evaluation at time point ^{n}t.

$^{mn}\Delta\sigma_{ij}$ stress component range between time points ^{m}t and ^{n}t.

$^{mn}\Delta\sigma_{11}$ stress range associated with the normal stress component in the 1-direction between time points ^{m}t and ^{n}t.

$^{mn}\Delta\sigma_{22}$ stress range associated with the normal stress component in the 2-direction between time points ^{m}t and ^{n}t.

$^{mn}\Delta\sigma_{33}$ stress range associated with the normal stress component in the 3-direction between time points ^{m}t and ^{n}t.

$^{mn}\Delta\sigma_{12}$ stress range associated with the shear stress component in the 1-direction between time points ^{m}t and ^{n}t.

$^{mn}\Delta\sigma_{13}$ stress range associated with the shear stress component in the 2-direction between time points ^{m}t and ^{n}t.

$^{mn}\Delta\sigma_{23}$ stress range associated with the shear stress component in the 3-direction between time points ^{m}t and ^{n}t.

^{m}t time point under consideration with the highest peak or lowest valley.

^{n}t time point under consideration that forms a range with time point ^{m}t.

^{m}N specified number of repetitions of the event associated with time point ^{m}t.

^{n}N specified number of repetitions of the event associated with time point ^{n}t.

X absolute value of the range (load or stress) under consideration using the Rainflow Cycle Counting Method.

Y absolute value of the adjacent range (load or stress) to previous X using the Rainflow Cycle Counting Method.

ANNEX 5.C

ALTERNATIVE PLASTICITY ADJUSTMENT FACTORS AND EFFECTIVE ALTERNATING STRESS FOR ELASTIC FATIGUE ANALYSIS

(NORMATIVE)

5.C.1 Scope

5.C.1.1 This Annex contains procedures for the determination of plasticity correction factors and effective alternating equivalent stress for elastic fatigue analysis. These procedures include a modified Poisson's ratio adjustment for local thermal and thermal bending stresses, a notch plasticity adjustment factor that is applied to thermal bending stresses, and a non-local plastic strain redistribution adjustment that is applied to all stresses except local thermal and thermal bending stresses. These procedures are an alternative to effective alternating stress calculations in Step 4 of paragraph 5.5.3.2.

5.C.2 Definitions

5.C.2.1 Thermal Bending Stress - Thermal bending stress is caused by the linear portion of the through-wall temperature gradient. Such stresses shall be classified as secondary stresses.

5.C.2.2 Local Thermal Stress - Local thermal stress is associated with almost complete suppression of the differential expansion and thus produces no significant distortion. Such stresses shall be considered only from the fatigue standpoint and are therefore classified as peak stresses. Examples of local thermal stresses are the stress in a small hot spot in a vessel wall, the non-linear portion of a through-wall temperature gradient in a cylindrical shell, and the thermal stress in a cladding material that has a coefficient of expansion different from that of the base metal. Local thermal stresses are characterized by having two principal stresses that are approximately equal.

5.C.3 Effective Alternating Stress for Elastic Fatigue Analysis

5.C.3.1 The effective total equivalent stress amplitude is used to evaluate the fatigue damage for results obtained from a linear elastic stress analysis. The controlling stress for the fatigue evaluation is the effective total equivalent stress amplitude, defined as one-half of the effective total equivalent stress range $\left(P_L + P_b + Q + F\right)$ calculated for each cycle in the loading histogram.

5.C.3.2 The following procedure shall be used to determine plasticity correction factors for elastic fatigue analysis and the effective alternating equivalent stress.

a) STEP 1 – At the point of interest, determine the following stress tensors and associated equivalent stresses at the start and end points (time points ${}^{m}t$ and ${}^{n}t$, respectively) for the k^{th} cycle counted in Step 2 of paragraph 5.5.3.2.

   1) Calculate the component stress range between time points ${}^{m}t$ and ${}^{n}t$ and compute an equivalent stress range due to primary plus secondary plus peak stress as given below.

$$\Delta\sigma_{ij,k} = {}^{m}\sigma_{ij,k} - {}^{n}\sigma_{ij,k} \tag{5.C.1}$$

$$\Delta S_{P,k} = \frac{1}{\sqrt{2}}\left[\begin{array}{l}\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \\ \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right)\end{array}\right]^{0.5} \tag{5.C.2}$$

2) Using the linearized stress results due to primary plus secondary stresses, compute the component stress range using Equation (5.C.1). Compute the equivalent stress range using Equation (5.C.2) and designate this quantity as $\Delta S_{n,k}$.

3) Determine the stress tensor due to local thermal and thermal bending stresses at the start and end points for the k^{th} cycle. It may be difficult to calculate the local thermal stress from stress distributions obtained from numerical methods. If this is the case, the procedure below can be used to calculate the local thermal and thermal bending stresses due to a non-linear temperature distribution. This method is based on calculating a thermal stress difference range associated with the linearized temperature distribution along the SCL for the time steps of interest. Consistent with that method, consider the distribution of the temperature from numerical method as a function of the local through thickness direction. The temperature distribution for each time step can be separated into three parts.

   i) A constant temperature equal to the average of the temperature distribution

$$T_{avg} = \frac{1}{t}\int_0^t T dx \tag{5.C.3}$$

   ii) The linearly varying portion of the temperature distribution

$$T_b = \frac{6}{t^2}\int_0^t T\left(\frac{t}{2} - x\right)dx \tag{5.C.4}$$

   iii) The non-linear portion of the temperature distribution

$$T_p = T - \left(T_{avg} + 2T_b / t\right) \tag{5.C.5}$$

By assuming full suppression of the differential expansion of the cross-section, the associated local thermal stress parallel to the surface for each time step may be calculated as given below where T_p is given by Equation (5.C.5).

$$\sigma_{ij,k}^{LT} = \frac{-E\alpha\left[T - \left(T_{avg} + 2T_b / t\right)\right]}{1-\nu} \qquad for \quad i = j = 1,2 \tag{5.C.6}$$

$$\sigma_{ij,k}^{LT} = 0 \qquad for \quad i \neq j \quad and \quad i = j = 3 \tag{5.C.7}$$

Using Equations (5.C.6) and (5.C.7), determine the local thermal component stress ranges using Equation (5.C.1) and designate this quantity as $\Delta\sigma_{ij,k}^{LT}$. The thermal bending component stress range, $\Delta\sigma_{ij,k}^{TB}$ is determined by linearizing the through-wall stress distribution due to thermal effects only.

4) Compute the equivalent stress ranges due to primary plus secondary plus peak stress minus the local thermal stress using Equations (5.C.8) and 5.C.9).

$$\Delta\sigma_{ij,k} = \left({}^{m}\sigma_{ij,k} - {}^{m}\sigma_{ij,k}^{LT}\right) - \left({}^{n}\sigma_{ij,k} - {}^{n}\sigma_{ij,k}^{LT}\right) \tag{5.C.8}$$

$$\left(\Delta S_{P,k} - \Delta S_{LT,k}\right) = \frac{1}{\sqrt{2}}\begin{bmatrix}\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \\ \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right)\end{bmatrix}^{0.5} \tag{5.C.9}$$

5) Compute the equivalent stress ranges due to local thermal plus thermal bending stress using Equations (5.C.10) and (5.C.11).

$$\Delta\sigma_{ij,k} = \left({}^{m}\sigma_{ij,k}^{TB} + {}^{m}\sigma_{ij,k}^{LT}\right) - \left({}^{n}\sigma_{ij,k}^{TB} + {}^{n}\sigma_{ij,k}^{LT}\right) \tag{5.C.10}$$

$$\left(\Delta S_{LT,k} + \Delta S_{TB,k}\right) = \frac{1}{\sqrt{2}}\begin{bmatrix}\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \\ \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right)\end{bmatrix}^{0.5} \tag{5.C.11}$$

6) If required, see Equation (5.C.32), compute the stress tensor due to non-thermal effects (all loadings except local thermal and thermal bending), $\sigma_{ij,k}^{NT}$, at the start and end points for the k^{th} cycle.

b) STEP 2 - Determine the Poisson's ratio adjustment, $K_{v,k}$ to adjust local thermal and thermal bending stresses for the k^{th} cycle based on the equivalent stress ranges in STEP 1 using the following equations, (S_{PS} is defined in paragraph 5.5.6.1):

$$K_{v,k} = 1.0 \qquad for\ \Delta S_{P,k} \le S_{PS} \tag{5.C.12}$$

$$K_{v,k} = 0.6\left[\frac{\left(\Delta S_{P,k} - S_{PS}\right)}{\left(\Delta S_{LT,k} + \Delta S_{TB,k}\right)}\right] + 1.0 \qquad for \quad \left|\begin{array}{l}\Delta S_{P,k} > S_{PS} \quad and \\ \left(\Delta S_{LT,k} + \Delta S_{TB,k}\right) > \left(\Delta S_{P,k} - S_{PS}\right)\end{array}\right. \tag{5.C.13}$$

$$K_{v,k} = 1.6 \qquad for \quad \left|\begin{array}{l}\Delta S_{P,k} > S_{PS} \quad and \\ \left(\Delta S_{LT,k} + \Delta S_{TB,k}\right) \le \left(\Delta S_{P,k} - S_{PS}\right)\end{array}\right. \tag{5.C.14}$$

c) STEP 3 – Determine the non-local plastic strain redistribution adjustment, $K_{nl,k}$ to adjust all stresses except local thermal and thermal bending for the k^{th} cycle. In these equations, the parameters m and n are defined in Table 5.13.

$$K_{nl,k} = 1.0 \qquad for \quad \Delta S_{n,k} \le S_{PS} \tag{5.C.15}$$

$$K_{nl,k} = 1.0 + \frac{(1-n)}{n(m-1)}\left(\frac{\Delta S_{n,k}}{S_{PS}} - 1\right) \qquad for \quad S_{PS} < \Delta S_{n,k} < mS_{PS} \tag{5.C.16}$$

$$K_{nl,k} = \frac{1}{n} \qquad for \qquad \Delta S_{n,k} \geq mS_{PS} \tag{5.C.17}$$

d) STEP 4 – Determine the notch plasticity adjustment factor, $K_{np,k}$ based on the equivalent stress ranges in STEP 1 to adjust thermal bending stresses to account for additional local strain concentration due to a geometric stress riser for the k^{th} cycle. In these equations, the parameters n are defined in Table 5.13.

For numerical results used directly:

$$K_{np,k} = 1.0 \qquad for \qquad \left(\Delta S_{P,k} - \Delta S_{LT,k}\right) \leq S_{PS} \tag{5.C.18}$$

$$K_{np,k} = \min\left[K_1,\ K_2\right] \qquad for \qquad \left(\Delta S_{P,k} - \Delta S_{LT,k}\right) > S_{PS} \tag{5.C.19}$$

$$K_1 = \left[\left(\frac{\Delta S_{P,k} - \Delta S_{LT,k}}{\Delta S_{n,k}}\right)^{\left(\frac{1-n}{1+n}\right)} - 1.0\right] \cdot \left[\frac{\left(\Delta S_{P,k} - \Delta S_{LT,k}\right) - S_{PS}}{\left(\Delta S_{P,k} - \Delta S_{LT,k}\right)}\right] + 1.0 \tag{5.C.20}$$

$$K_2 = \frac{K_{nl,k}}{K_{v,k}} \tag{5.C.21}$$

For numerical results that are adjusted with a stress concentration factor (SCF):

$$K_{np,k} = 1.0 \qquad for \qquad \left(\Delta S_{n,k} \cdot SCF\right) \leq S_{PS} \tag{5.C.22}$$

$$K_{np,k} = \min\left[K_1,\ K_2\right] \qquad for \qquad \left(\Delta S_{n,k} \cdot SCF\right) > S_{PS} \tag{5.C.23}$$

$$K_1 = \left[\left(SCF\right)^{\left(\frac{1-n}{1+n}\right)} - 1.0\right] \cdot \left[\frac{\left(\Delta S_{n,k} \cdot SCF\right) - S_{PS}}{\left(\Delta S_{n,k} \cdot SCF\right)}\right] + 1.0 \tag{5.C.24}$$

$$K_2 = \frac{K_{nl,k}}{K_{v,k}} \tag{5.C.25}$$

Note that the SCF and $K_{np,k}$ values may be dependent upon the component stress direction.

e) STEP 5 – Apply the plasticity adjustment factors to the component stresses at the start and end points for the k^{th} cycle.

1) Compute the component stresses including plastic Poisson's ratio and notch plasticity adjustments as given below for time points ${}^{m}t$ and ${}^{n}t$.

For numerical results used directly:

$$\left(\sigma_{ij}^{LT}\right)_{adj} = \sigma_{ij,k}^{LT} \cdot K_{v,k} \tag{5.C.26}$$

$$\left(\sigma_{ij}^{TB}\right)_{adj} = \sigma_{ij,k}^{TB} \cdot K_{v,k} \cdot K_{np,k} + \sigma_{ij,k}^{TB} \cdot \left(SCF_{NUM} - 1\right) \cdot K_{np,k} \tag{5.C.27}$$

For numerical results that are adjusted with a stress concentration factor (SCF):

$$\left(\sigma_{ij}^{LT}\right)_{adj} = \sigma_{ij,k}^{LT} \cdot K_{v,k} \cdot SCF_{LT} \tag{5.C.28}$$

$$\left(\sigma_{ij}^{TB}\right)_{adj} = \sigma_{ij,k}^{TB} \cdot K_{v,k} \cdot K_{np,k} \cdot SCF + \sigma_{ij,k}^{TB} \cdot \left(SCF_{NUM} - 1\right) \cdot K_{np,k} \tag{5.C.29}$$

2) Compute the component stresses including non-local plastic strain redistribution adjustment as given below for time points ${}^{m}t$ and ${}^{n}t$.

For numerical results used directly:

$$\left(\sigma_{ij}^{NT}\right)_{adj} = \left[\sigma_{ij,k} - \sigma_{ij,k}^{TB}\left(SCF_{NUM} - 1\right)\right] \cdot K_{np,k} \tag{5.C.30}$$

$$SCF_{NUM} = \frac{\left(\Delta S_{p,k} - \Delta S_{LT,k}\right)}{\Delta S_{n,k}} \tag{5.C.31}$$

For numerical results that are adjusted with a stress concentration factor (SCF):

$$\left(\sigma_{ij}^{NT}\right)_{adj} = \sigma_{ij,k}^{NT} \cdot K_{nl,k} \cdot SCF \tag{5.C.32}$$

f) STEP 6 – Compute the adjusted component stress ranges between time points ${}^{m}t$ and ${}^{n}t$ as given below.

$$\left(\Delta\sigma_{ij,k}\right)_{adj} = \left\{ \begin{array}{l} {}^{m}\left[\left(\sigma_{ij}^{LT}\right)_{adj} + \left(\sigma_{ij}^{NT}\right)_{adj} + \left(\sigma_{ij}^{TB}\right)_{adj}\right] - \\ {}^{n}\left[\left(\sigma_{ij}^{LT}\right)_{adj} + \left(\sigma_{ij}^{NT}\right)_{adj} + \left(\sigma_{ij}^{TB}\right)_{adj}\right] \end{array} \right\} \tag{5.C.33}$$

g) STEP 7 – Compute the effective equivalent stress range using the adjusted component stress ranges from STEP 6 and Equation (5.C.2). Designate the adjusted effective equivalent stress range as $\left(\Delta S_{P,k}\right)_{adj}$.

h) STEP 8 – Compute the effective alternating equivalent stress for the k^{th} cycle as given below.

$$S_{alt,k} = 0.5\left(\Delta S_{P,k}\right)_{adj} \tag{5.C.34}$$

5.C.4 Nomenclature

α thermal expansion coefficient of the material at the point under consideration, evaluated at the mean temperature of the k^{th} cycle.

$\Delta S_{n,k}$ primary plus secondary equivalent stress range for the k^{th} cycle.

$\Delta S_{P,k}$ range of primary plus secondary plus peak equivalent stress for the k^{th} cycle.

$\Delta S_{LT,k}$ primary plus secondary plus peak equivalent stress range due to local thermal effects for the k^{th} cycle.

$\Delta S_{TB,k}$ primary plus secondary plus peak equivalent stress range due to thermal bending effects for the k^{th} cycle.

$\Delta S_{NT,k}$ primary plus secondary plus peak equivalent stress range due to non-thermal effects for the k^{th} cycle.

$\left(\Delta S_{P,k}\right)_{adj}$ adjusted range of primary plus secondary plus peak equivalent stress, including non-local strain redistribution, notch plasticity, and plastic Poisson's ratio adjustments for the k^{th} cycle.

$\Delta\sigma_{ij,k}$ stress component range between time points ${}^{m}t$ and ${}^{n}t$ for the k^{th} cycle.

$\Delta\sigma_{ij,k}^{LT}$ stress component range due to local thermal stress between time point ${}^{m}t$ and ${}^{n}t$ for the k^{th} cycle.

$\Delta\sigma_{ij,k}^{TB}$ stress component range due to thermal bending stress between time points ${}^{m}t$ and ${}^{n}t$ for the k^{th} cycle.

adjusted stress tensor, including non-local strain redistribution, notch plasticity, and plastic Poisson's ratio adjustments at the location and time point under evaluation for the k^{th} cycle.

E Young's Modulus of the material evaluated at the mean temperature of the cycle.

K_1 parameter used to compute $K_{np,k}$.

K_2 parameter used to compute $K_{np,k}$..

$K_{nl,k}$ non-local strain redistribution adjustment factor for the k^{th} cycle.

$K_{np,k}$ notch plasticity adjustment factor for the k^{th} cycle.

$K_{\nu,k}$ plastic Poisson's ratio adjustment factor for the k^{th} cycle.

m material constant used for the non-local strain redistribution adjustment factor, per Table 5.6.

n material constant used for the non-local strain redistribution adjustment factor, per Table 5.6.

ν Poisson's ratio.

$S_{alt,k}$ alternating equivalent stress for the k^{th} cycle.

S_{PS} allowable limit on the primary plus secondary stress range.

SCF stress concentration factor.

SCF_{LT} stress concentration factor applicable for local thermal stress.

SCF_{NUM} stress concentration factor determined from the numerical model.

$\sigma_{ij,k}^{LT}$ stress tensor due to local thermal stress at the location and time point under evaluation for the k^{th} cycle.

$\sigma_{ij,k}^{NT}$ stress tensor due to non-thermal stress at the location and time point under evaluation for the k^{th} cycle.

$\sigma_{ij,k}^{TB}$ stress tensor due to thermal bending stress due to the linearly varying portion of the temperature distribution at the location and time point under evaluation for the k^{th} cycle.

${}^{m}\sigma_{ij,k}$ stress tensor at the point under evaluation at time point ${}^{m}t$ for the k^{th} cycle.

${}^{n}\sigma_{ij,k}$ stress tensor at the point under evaluation at time point ${}^{n}t$ for the k^{th} cycle.

${}^{m}\sigma_{ij,k}^{LT}$ stress tensor due to local thermal stress at the location under evaluation at time point ${}^{m}t$ for the k^{th} cycle.

${}^{n}\sigma_{ij,k}^{LT}$ stress tensor due to local thermal stress at the location under evaluation at time point ${}^{n}t$ for the k^{th} cycle.

${}^{m}\sigma_{ij,k}^{TB}$ stress tensor due to thermal bending stress at the location under evaluation at time point ${}^{m}t$ for the k^{th} cycle.

${}^{n}\sigma_{ij,k}^{TB}$ stress tensor due to thermal bending stress at the location under evaluation at time point ${}^{n}t$ for the k^{th} cycle.

$\left(\sigma_{ij}^{LT}\right)_{adj}$ adjusted stress tensor due to local thermal stress at the location and time point under evaluation for the k^{th} cycle.

$\left(\sigma_{ij}^{NT}\right)_{adj}$ adjusted stress tensor due to non-thermal stress at the location and time point under evaluation for the k^{th} cycle.

$\left(\sigma_{ij}^{TB}\right)_{adj}$ adjusted stress tensor due to thermal bending stress at the location and time point under evaluation for the k^{th} cycle.

t wall thickness.

${}^{m}t$ time point under consideration with the highest peak or lowest valley.

${}^{n}t$ time point under consideration that forms a range with time point ${}^{m}t$.

T temperature distribution.

T_{avg} average temperature component of temperature distribution T.

T_b equivalent linear temperature component of temperature distribution T.

T_p peak temperature component of temperature distribution T.

x position through the wall thickness.

z local coordinate for the temperature distribution.

ANNEX 5.D

STRESS INDICES

(NORMATIVE)

5.D.1 General

5.D.1.1 In lieu of a detailed stress analysis, stress indices may be used to determine peak stresses around a nozzle opening.

5.D.1.2 The term stress index, is defined as the numerical ratio of the stress components σ_t, σ_n, and σ_r under consideration to the computed membrane hoop stress in the unreinforced vessel material; however, the material which increases the thickness of a vessel wall locally at the nozzle shall not be included in the calculation of these stress components. These stress directions are defined in Figure 5.D.1. When the thickness of the vessel wall is increased over that required to the extent provided hereinafter, the values of r_1 and r_2 in Figure 5.D.2 shall be referred to the thickened section.

5.D.1.3 The stress indices in these tables provide only the maximum stresses at certain general locations due to internal pressure. In the evaluation of stresses in or adjacent to vessel openings and connections, it is often necessary to consider the effect of stresses due to external loadings or thermal stresses. In such cases, the combined stress at a given point may be determined by superposition. In the case of combined stresses due to internal pressure and nozzle loading, the maximum stresses for a given location should be considered as acting at the same point and added algebraically unless positive evidence is available to the contrary.

5.D.2 Stress Indices for Radial Nozzles

5.D.2.1 The stress indices for radial nozzles in spherical shells and spherical portions of formed heads in Table 5.D.1 and for nozzles in cylindrical shells in Table 5.D.2 may be used if all of the items listed below are true.

a) The opening is for a circular nozzle whose axis is normal to the vessel wall. If the axis of the nozzle makes an angle θ with the normal to the vessel wall, an estimate of the σ_n index on the inside may be obtained from the equations shown below provided $d_{ni}/D_i \le 0.15$. In these equations, K_1 is the σ_n stress index for a radial connection and K_2 is the σ_n stress index for a non-radial connection.

$$K_2 = K_1\left(1+2\sin^2\theta\right) \qquad (\textit{for hillside nozzles in spheres or cylinders}) \tag{5.D.1}$$

$$K_2 = K_1\left(1+\left(\tan\theta\right)^{4/3}\right) \qquad (\textit{for lateral nozzles in cylinders}) \tag{5.D.2}$$

b) The arc distance measured between the centerlines of adjacent nozzles along the inside surface of the shell is not less than three times the sum of their inside radii for openings in a head or along the longitudinal axis of a shell and is not less than two times the sum of their radii for openings along the circumference of a cylindrical shell.

c) For nozzles in cylindrical shells, the following dimensional limitations shown below are satisfied. In addition, the total nozzle reinforcement area on the transverse plane of the nozzle including any outside of the reinforcement limits shall not exceed 200% of that required for the longitudinal plane unless a tapered transition section is incorporated into the reinforcement and the shell.

$$10 \le \frac{D_i}{t} \le 100 \tag{5.D.3}$$

$$\frac{d_{ni}}{D_i} \le 0.50 \tag{5.D.4}$$

$$\frac{d_{ni}}{\sqrt{\dfrac{D_i t_n r_2}{t}}} \le 1.50 \tag{5.D.5}$$

d) For nozzles in spherical shells, the following dimensional limitations shown below are satisfied. In addition, at least 40% of the reinforcement is located on the outside surface of the nozzle-shell juncture.

$$10 \le \frac{D_i}{t} \le 100 \tag{5.D.6}$$

$$\frac{d_{ni}}{D_i} \le 0.50 \tag{5.D.7}$$

$$\frac{d_{ni}}{\sqrt{D_i t}} \le 0.80 \tag{5.D.8}$$

e) For nozzles in cylindrical and spherical shells, the following local geometry details shall be satisfied.

1) The inside corner radius r_1 (see Figure 5.D.2) is one-eighth to one-half of the shell thickness t.

2) The outer corner radius r_2 (see Figure 5.D.2) is large enough to provide a smooth transition between the nozzles and the shell. In addition, for nozzle diameters greater than 1.5 shell thicknesses in cylindrical shells and 2:1 ellipsoidal heads, or for nozzle diameters greater than 3 shell thicknesses in spherical shells, the value of r_2 shall satisfy the following:

$$r_2 \ge \max\left[\sqrt{2rt_n},\ \frac{t}{2}\right] \tag{5.D.9}$$

3) The value of r_3 shall satisfy the following:

$$r_3 \ge \max\left[\sqrt{rt_p},\ \frac{t_n}{2}\right] \tag{5.D.10}$$

5.D.2.2 Stress indices for radial nozzles in spherical shells and spherical portions of formed heads and nozzle in cylindrical shells may be developed by stress analysis techniques consistent with the elastic design analysis methods of Part 5, or obtained for other sources.

5.D.3 Stress Indices for Laterals

5.D.3.1 The stress indices for laterals in cylindrical shells in Table 5.D.3 may be used when all of the following conditions are satisfied:

a) The angle θ equals 45 deg; these indices may be used for angles of θ less than 45 degrees (see Figure 5.D.3).

b) The nozzle is of circular cross section and has an axis that intersects the axis of the cylindrical vessel.

c) The design of the nozzle reinforcement is in accordance with the applicable rules of paragraph 4.5.

d) The following dimensional ratios are satisfied.

$$\frac{D_i}{t} \le 40.0 \tag{5.D.11}$$

$$\frac{d_{ni}}{D_i} \le 0.5 \tag{5.D.12}$$

$$\frac{d_{ni}}{\sqrt{D_i t}} \le 3.0 \tag{5.D.13}$$

e) The nominal pressure membrane stress to be used with the pressure indices is determined using the following equations.

$$\sigma_p = \frac{P\left(D_i + t\right)}{2t} \qquad (for\ Regions\ 1\ and\ 2) \tag{5.D.14}$$

$$\sigma_p = \frac{P\left(d_{ni} + t_p\right)}{2t_p} \qquad (for\ Region\ 3) \tag{5.D.15}$$

5.D.3.2 Stress indices for laterals in cylindrical shells may be developed by stress analysis techniques consistent with the elastic design analysis methods of Part 5, or obtained for other sources.

5.D.4 Nomenclature

D_i inside vessel diameter.

d_{ni} inside nozzle diameter.

p pressure.

r_1 local nozzle radius, see Figure 5.D.2.

r_2 local nozzle radius, see Figure 5.D.2.

r_3 local nozzle radius, see Figure 5.D.2.

σ_r stress component normal to the boundary of the section.

σ_t stress component in the plane of the section under consideration and parallel to the boundary of the section.

σ_n stress component normal to the plane of the section (ordinarily the circumferential stress around the hole in the shell.

t minimum wall thickness in the region under consideration, or the thickness of the vessel, as applicable.

t_p thickness of perforated plate, or the thickness of the pipe portion of a nozzle, as applicable.

θ angle between the axis of the nozzle and the normal to the vessel.

5.D.5 Tables

Table 5.D.1 – Stress Indices For Nozzles In Spherical Shells And Portions Of Formed Heads

Stress	Inside Corner	Outside Corner
σ_n	2.0	2.0
σ_t	–0.2	2.0
σ_r	$-\frac{2t}{R}$	0.0
σ	2.2	2.0

Table 5.D.2 – Stress Indices For Nozzles In Cylindrical Shells

Material	Longitudinal Plane		Transverse Plane	
	Inside Corner	Outside Corner	Inside Corner	Outside Corner
σ_n	3.1	1.2	1.0	2.1
σ_t	–0.2	1.0	–0.2	2.6
σ_r	$-\frac{t}{R}$	0.0	$-\frac{t}{R}$	0.0
σ	3.3	1.2	1.2	2.6

Table 5.D.3 – Stress Indices For Laterals

Load	Stress	Region 1		Region 2		Region 3	
		Inside (1)	Outside (1)	Inside	Outside	Inside	Outside
Pressure	$\sigma_{\max}$	5.5	0.8	3.3	0.7	1.0	1.0
	S	5.75	0.8	3.5	0.75	1.2	1.1
In-Plane Branch Moment, M_B	$\sigma_{\max}$	0.1	0.1	0.5	0.5	1.0	1.6
	S	0.1	0.1	0.5	0.5	1.0	1.6
Vessel Moments, M_R or M_{RT}	$\sigma_{\max}$	2.4	2.4	0.6	1.8	0.2	0.2
	S	2.7	2.7	0.7	2.0	0.3	0.3
Out-Of-Plane Branch Moment, M_{BT}	$\sigma_{\max}$	0.13	NA	0.06	NA	NA	2.5 (2)
	S	0.22	NA	0.07	NA	NA	2.5 (2)

Notes:

1. Inside/outside refers to inside corner (pressure side)/outside fillet and in the plane of symmetry as shown in Figure 5.D.3
2. Maximum stress/stress intensity in Region 3 for transverse moment M_{BT} occurs 90° away from in-plane moment.

5.D.6 Figures

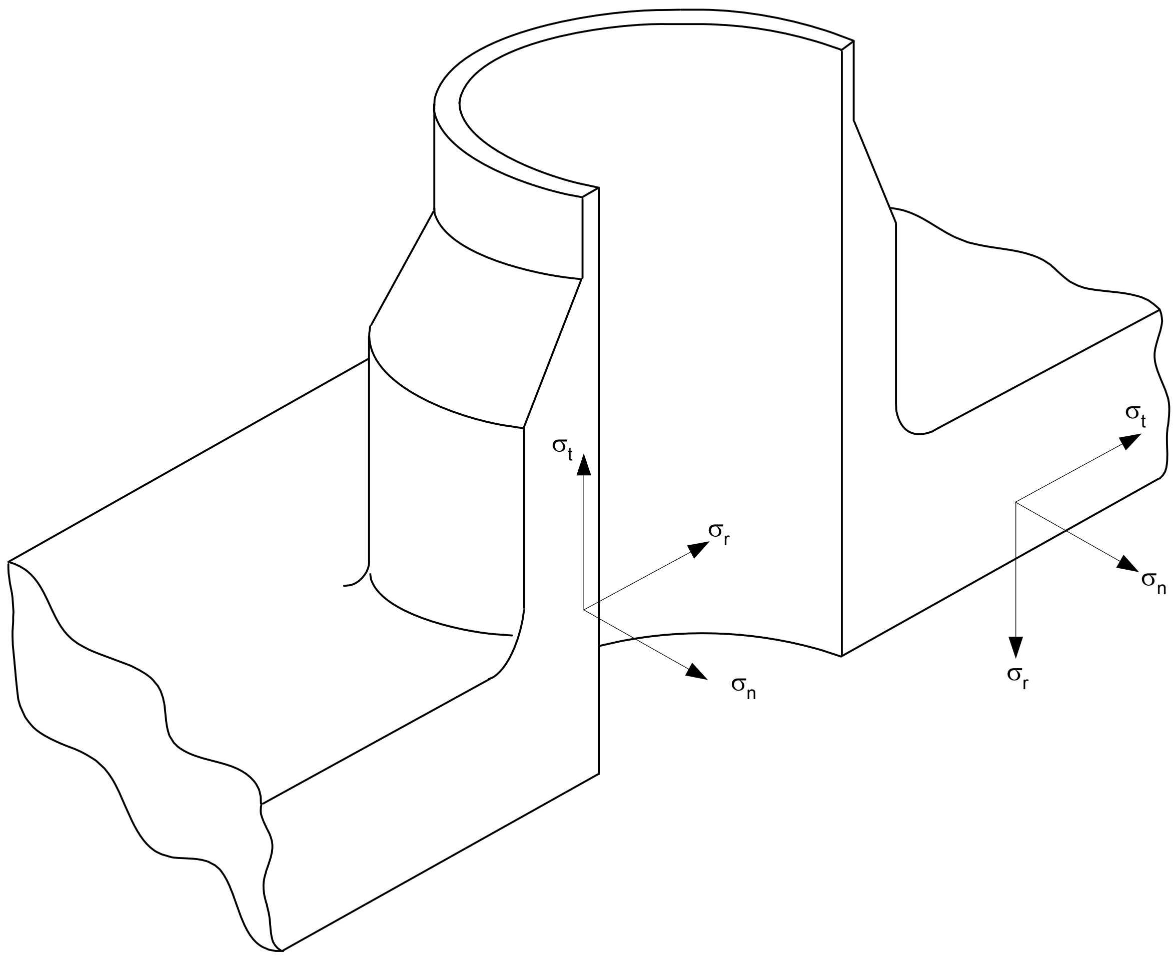


Figure 5.D.1
Direction of Stress Components

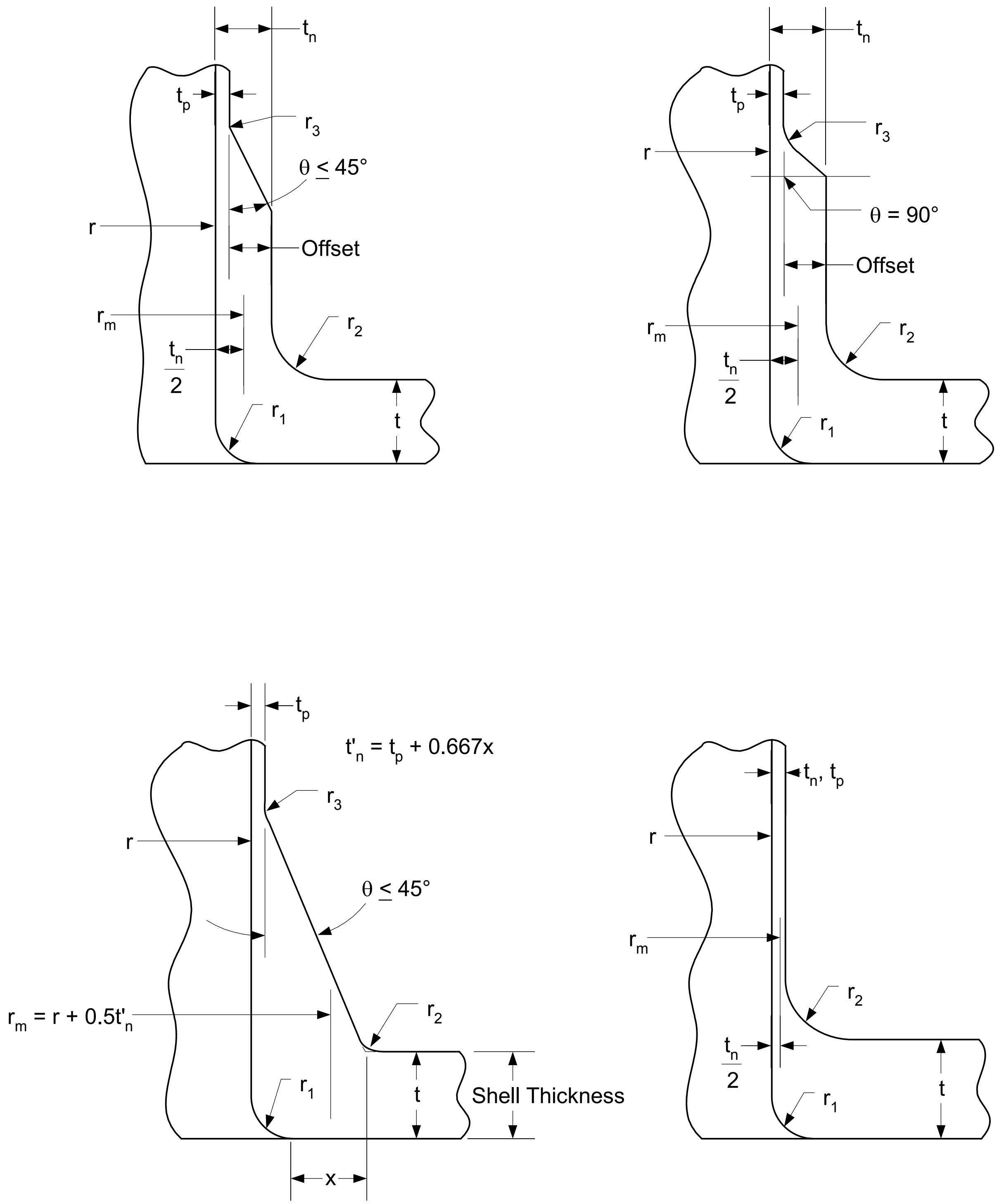


Figure 5.D.2
Nozzle Nomenclature and Dimensions

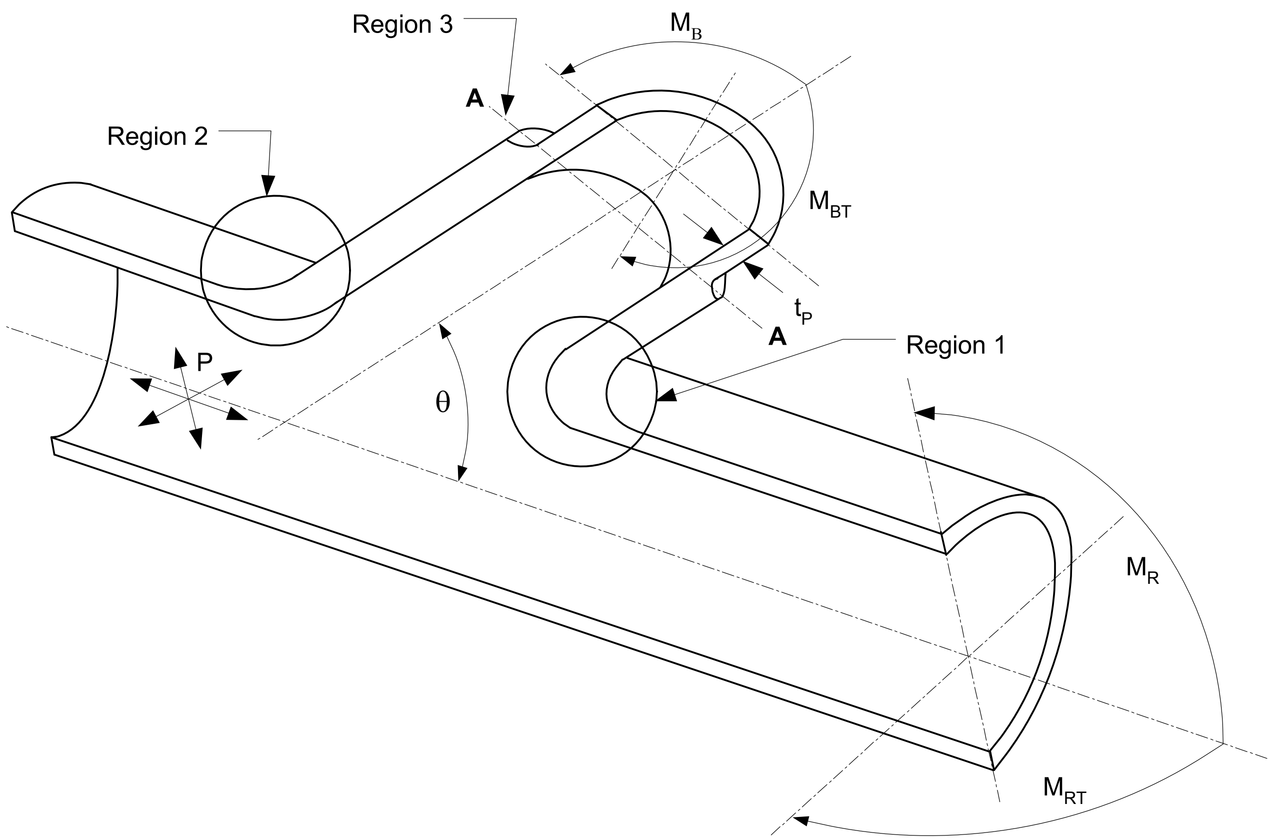


Figure 5.D.3
Nomenclature and Loading for Laterals

ANNEX 5.E

DESIGN METHODS FOR PERFORATED PLATES BASED ON ELASTIC STRESS ANALYSIS

(NORMATIVE)

5.E.1 Overview

5.E.1.1 This paragraph contains a method of analysis for flat perforated plates subjected to applied loads or loadings resulting from structural interaction with adjacent members. This method applies to perforated plates that satisfy the following conditions:

a) The holes are in an equilateral triangular or square penetration pattern.

b) The holes are circular and the axis of the hole is perpendicular to the surface of the plate.

c) There are 19 or more holes.

d) The effective ligament efficiency satisfies the following criteria in paragraph 5.E.4.

5.E.1.2 Rolling or pressure expansion of tubes into a tubesheet will result in compression on the inside diameter of the tubesheet perforation and the tube itself. Thermal transients that occur during operation may result in thermal differential between the tube and the tubesheet resulting in a loss of the compression obtained during fabrication. For tubes that are also welded, such thermal cycling can cause fatigue of the weld joining the tube to the tubesheet. For tubes that are not welded, thermal cycling can cause the tube to loose contact resulting in leakage. A transient thermal and stress analysis can determine the magnitude of cyclic stress in the tube weld. For small ligaments, the pressure expansion of tubes can cause significant ligament deformation and stresses.

5.E.2 Stress Analysis of the Equivalent Solid Plate

5.E.2.1 The analysis method for perforated plates presented in this paragraph is based on the concept of an effective solid plate. In this method, the material properties of the actual perforated plate are replaced by material properties of an effective solid plate that approximate the stiffness of the actual perforated plate. A stress analysis is then performed where the effective solid plate is used in the model that is geometrically similar to the perforated plate, and subject to the actual loading conditions. After completion of the stress analysis, the values of the stresses in the actual perforated plate are obtained by applying stress multiplying factors to the stress results computed for the effective solid plate.

5.E.2.2 An accurate model of the overall tubesheet behavior may be achieved by employing the concept of an equivalent elastic material of anisotropic properties. For triangular penetration patterns the in-plane behavior of the tubesheet is isotropic and the anisotropy of the equivalent material must only be considered for stresses in the thickness direction. The tubesheet can be analyzed using an axisymmetric solid numerical analysis with the effective elastic matrix $[E]$ to simulate the anisotropic behavior. For square penetration patterns the equivalent material properties depend on the orientation of the loading with respect to the symmetry axes of the pattern. The tubesheet can be analyzed using an axisymmetric solid numerical analysis with the effective average elastic matrix $[E]$ for in-plane loading to simulate the anisotropic behavior. When the tubesheet loading is not axisymmetric (e.g. channels with divider plates), a three-dimensional solid numerical analysis may be required.

5.E.3 Stiffness Effects of the Tubes

5.E.3.1 The in-plane stiffening effect of the tubes in the perforations may be included in the analysis. The extent to which the tubes stiffen the perforated plate depends on the materials, the manufacturing processes, operating conditions, and degree of corrosion. This stiffening effect may be included in the calculations by including part or all of the tube walls in the ligament efficiency used to obtain the effective elastic constants of the plate. Such stiffening may either increase or decrease stresses in the plate itself and in the attached shells.

5.E.3.2 Where applicable, the stiffening effect resulting from the staying action of the tubes should be incorporated into the numerical analysis. For fixed-fixed type exchangers, the axial stiffness of the tubes shall be included in the analysis. When including the tube stiffness, the difference in axial strain due to pressure between the shell and tubes (Poisson's effect) as well as differential thermal expansion between tubes and shell should also be included in the numerical analysis.

5.E.4 Effective Material Properties for the Equivalent Solid Plate

5.E.4.1 The effective elastic constants of the perforated plate are determined as a function of the effective ligament efficiency μ^* that is computed in accordance with paragraph 4.18.6.4.

5.E.4.2 If the hole pattern is irregular or has isolated thin ligaments in a uniform hole pattern, the calculated elastic constants shall represent the average ligament size over the entire modeled plate in order to capture the global response. The stress multipliers used for the fatigue assessment of paragraph 5.E.7.2 shall be computed considering the appropriate ligament efficiency for the location of interest.

5.E.4.3 The effective elastic constants of the perforated plate material are determined based on the hole pattern and thickness of the plate. Either one of the following two options may be used in the analysis.

a) Option A – The effective elastic constants are provided below. Elastic constants are provided for the effective elastic modulus and effective Poisson's ratio. A value for the effective shear modulus, which is required in the analysis method in this annex, is not provided. The effective shear modulus shall be evaluated using Option B.

   1) Triangular Hole Pattern – The effective elastic modulus and effective Poison's ratio are provided in Tables 5.E.1 and 5.E.2, respectively.

   2) Square Hole Pattern – The effective elastic modulus and effective Poisson's ratio are provided in Tables 5.E.3 and 5.E.4, respectively. The effective elastic modulus and Poisson's ratio represent an equivalent isotropic value considering the values of the pitch and diagonal properties.

   3) Range of Applicability – The range of applicability in terms of the effective tube pitch is $0.1 \le \mu^* \le 1.0$. The effective elastic constants E^* and ν^* can be used for in-plane loading and bending loading of all values of h/p.

   4) The value of the in-plane shear modulus is given by Equation (5.E.1). The value of the out-of-plane shear modulus G_z^* shall be computed using Option B.

$$G^* = \frac{E^*}{2\left(1+\upsilon^*\right)} \tag{5.E.1}$$

b) Option B – The effective elastic constants for plane stress and generalized plane strain are provided below.

1) Triangular Hole Pattern – The effective elastic modulus and effective Poison's ratio are provided in Table 5.E.5

2) Square Hole Pattern – The effective elastic modulus and effective Poisson's ratio for the pitch and diagonal directions are provided in Tables 5.E.6 and 5.E.7, respectively.

3) Effects Of Poisson's Ratio – The effective material constants in Tables 5.E.5, 5.E.6, and 5.E.7 are based on a Poisson's ratio of $\nu = 0.3$. The material constants can be determined for other values of Poisson's ratio using the following procedure.

i) For a given value of μ^*, determine E^* and ν^* using the equations in Tables 5.E.5, 5.E.6, and 5.E.7, as applicable. These values correspond to $\nu = 0.3$.

ii) Determine F_0 using the following equation:

$$F_0 = \frac{E^*}{E} \tag{5.E.2}$$

iii) Determine F_1 using the following equation:

$$F_1 = \nu^* - 0.3F_0 \tag{5.E.3}$$

iv) The value of ν^* for any value of the Poisson's ratio, ν, is:

$$\nu^*(\nu) = F_1 + \nu F_0 \tag{5.E.4}$$

v) With the value of $\nu^*(\nu)$ determined above, the effective elastic modulus and shear modulus can be determined for any value of the Poisson's ratio by using the equations in Tables 5.E.5, 5.E.6, and 5.E.7, as applicable.

4) Range of Applicability – The range of applicability in terms of the effective tube pitch is $0.1 \le \mu^* \le 1.0$. The range of applicability in terms of h/p is:

i) The effective elastic constants $\overline{E}^*$ and $\overline{\nu}^*$ can be used for in-plane loading and bending loading when $h/p \ge 2.0$.

ii) The effective elastic constants E^* and ν^* can be used for in-plane loading when $h/p < 2.0$. The effective constants in Option A can be used for bending loading when $h/p < 2.0$.

iii) The effective elastic constants G^*, G_p^*, G_d^*, and G_z^* can be used for in-plane loading and bending loading for all values of h/p.

iv) The effective elastic constants for plane stress shall be used when $h/p < 2$, and the effective elastic constants for generalized plane strain shall be used when $h/p \ge 2$.

c) Option C – The effective elastic constants for the geometry being evaluated can be derived by stress analysis. In this case, the tube layout geometry is modeled and loads are applied to introduce normal and shear stresses. The effective elastic constants can be derived from the stress and strain results from the stress analysis.

5.E.4.4 The elasticity matrix $[E]$ for the triangular and square patterns that are used in the stress analysis of the effective solid plate are provided in Tables 5.E.8 and 5.E.9, respectively.

5.E.5 Pressure Effects in Tubesheet Perforations

5.E.5.1 The effect of pressure in the perforations is not directly included in the stress analyses of the effective solid plate. Pressure in the tubesheet perforations tends to expand the perforated portion of the plate. This expansion is resisted by the solid rim and by the ligaments themselves. The effect of this pressure on the local stresses in the ligaments can be included in the stress analysis results by use of the superposition principle.

5.E.5.2 In order to include the effect of pressure, an additional analysis needs to be performed. In this analysis, the boundary conditions are shown in Figure 5.E.2 (c). These boundary conditions can be achieved by superposing the results from the boundary conditions shown in Figures 5.E.2 (a) and 5.E.2 (b), respectively. The solution for the hydrostatic compression boundary condition case, see Figure 5.E.2 sketch (a) is $\sigma_r^* = \sigma_\theta^* = \sigma_z^* = -p_s$ for tubes welded on the tubeside or $\sigma_r^* = \sigma_\theta^* = \sigma_z^* = -p_t$ for tubes welded on the shellside. The solution for the boundary conditions of Figure 5.E.2 (b) can be obtained from the stress analyses considering the loading conditions shown.

5.E.5.3 A further modification to the boundary conditions used in paragraph 5.E.2 can be used to obtain more exact results. The modified boundary conditions are shown in Figure 5.E.3.

5.E.5.4 The stress results obtained using the boundary conditions shown in Figures 5.E.2 and Figure 5.E.3 must be superposed before P_m, $P_L + P_b$, and $P_L + P_b + Q$ stresses are evaluated and before transforming the cylindrical coordinate stresses to the local x-y-z coordinate system for computing $P_L + P_b + Q + F$ stresses.

5.E.6 Protection Against Plastic Collapse

5.E.6.1 Uniform Hole Pattern – The following equations shall be used for a perforated plate with a uniform hole pattern. The equivalent stresses P_m, $P_L + P_b$, and $P_L + P_b + Q$ are determined from the in-plane stress components determined by stress analysis of the equivalent solid plate. The loads to be considered in the design shall include, but not be limited to, those given in Table 4.1.1. The load combinations that shall be considered for each loading condition shall include, but not be limited to those given in Table 4.1.2.

a) General Primary Membrane Equivalent Stress (P_m) – evaluate the through thickness stress distribution from the numerical analysis of the equivalent plate for mechanical plus pressure loading. The membrane stress is determined by linearization of the through thickness stress distribution. The maximum stress intensity, P_m, shall satisfy the following:

$$\frac{P_m}{\mu^*} \le S_m \tag{5.E.5}$$

b) Primary Membrane (General or Local) Plus Primary Bending Equivalent Stress $(P_L + P_B)$ – determine the linearized surface stresses from the numerical analysis of the equivalent plate for mechanical plus pressure loading. The maximum stress intensity, $P_L + P_b$, shall satisfy the following:

$$\frac{\left(P_L+P_b\right)K_{PS}}{\mu^*}\leq 1.5S_m \tag{5.E.6}$$

where

$$K_{PS}=\left[\frac{1.07802-0.342503\beta+1.50452\beta^2}{1-0.0706632\beta+1.15182\beta^2+0.158343\beta^3}\right]^{0.5} \tag{5.E.7}$$

$$\beta=\frac{\sigma_r^*}{\sigma_\theta^*}\quad or\quad \frac{\sigma_\theta^*}{\sigma_r^*}\quad where\ \ -1\leq\beta\leq 1 \tag{5.E.8}$$

5.E.6.2 Primary Membrane (General or Local) Plus Primary Bending Plus Secondary Equivalent Stress (P_L+P_B+Q) – if a fatigue analysis is required, determine the linearized surface stresses from the numerical analysis of the equivalent plate for mechanical, pressure, and thermal loading. The maximum primary plus secondary equivalent stress, P_L+P_B+Q, shall satisfy the following where K_{PS} is computed using Equation (5.E.7).

$$\frac{(P_L+P_B+Q)K_{PS}}{\mu^*}\leq S_{PS} \tag{5.E.9}$$

5.E.7 Protection Against Cyclic Loading

5.E.7.1 Stress Components From The Equivalent Plate – The stress components σ_x^*, σ_y^*, σ_z^*, τ_{xy}^*, τ_{xz}^*, and τ_{yz}^*, are determined from the stress analysis of the equivalent solid plate.

a) Triangular Hole Pattern – For material constants evaluated using Option A, B, or C (see paragraph 5.E.4.3), the perforated plate displacements and stresses are minimally affected by the orientation with respect to the hole pattern. Therefore, the numerical stress results can be used directly in the evaluation.

b) Square Hole Pattern – The determination of the stress results depends on the material property model.

    1) For material constants evaluated using Option A or Option C (see paragraph 5.E.4.3), the perforated plate displacements and stresses are minimally affected by the orientation with respect to the hole pattern. Therefore, the numerical stress results can be used directly in the evaluation.

    2) For material constants evaluated using Option B, the square pattern displacements do not dependent strongly on the orientation with respect to the hole pattern; therefore, the axisymmetric approximation provides sufficiently accurate strain results. However, the stress components have directional dependence with the stiffness properties. Therefore, the stress components shall be determined using the equations in Table 5.E.10 where the elasticity matrix is dependent on the orientation of the hole pattern with respect to the equivalent plate cylindrical coordinate system. Using the axisymmetric strains in the plate, the non-axisymmetric nominal stresses in the pitch and diagonal directions for the square pattern perforations can be obtained from the equations below:

$$\left(\sigma_r^*\right)_p=\left(\sigma_r^*\right)_a-\frac{1}{2}\left(G_p^*-G_d^*\right)\left(\varepsilon_r^*-\varepsilon_\theta^*\right) \tag{5.E.10}$$

$$\left(\sigma_{\theta}^{*}\right)_{p}=\left(\sigma_{\theta}^{*}\right)_{a}+\frac{1}{2}\left(G_{p}^{*}-G_{d}^{*}\right)\left(\varepsilon_{r}^{*}-\varepsilon_{\theta}^{*}\right) \tag{5.E.11}$$

$$\left(\sigma_{r}^{*}\right)_{d}=\left(\sigma_{r}^{*}\right)_{a}+\frac{1}{2}\left(G_{p}^{*}-G_{d}^{*}\right)\left(\varepsilon_{r}^{*}-\varepsilon_{\theta}^{*}\right) \tag{5.E.12}$$

$$\left(\sigma_{\vartheta}^{*}\right)_{d}=\left(\sigma_{\theta}^{*}\right)_{a}-\frac{1}{2}\left(G_{p}^{*}-G_{d}^{*}\right)\left(\varepsilon_{r}^{*}-\varepsilon_{\theta}^{*}\right) \tag{5.E.13}$$

$$\sigma_{z}^{*}=\left(\sigma_{z}^{*}\right)_{a} \tag{5.E.14}$$

$$\sigma_{rz}^{*}=\left(\sigma_{rz}^{*}\right)_{a} \tag{5.E.15}$$

c) For both the triangular and square pattern perforated plates, the hydrostatic compression stress must be superposed prior to transformation to the x-y-z Cartesian coordinate system (see Figure 5.E.4 or 5.E.5) for computing the peak stresses.

5.E.7.2 Stress Components For Fatigue Assessment – The stress components for the fatigue assessment are computed using the following equations.

a) Stress Results In The Perforated Region Of The Plate – the stress multipliers K_x, K_y, K_{xy}, K_{xz}, and K_{yz} for triangular and square hole patterns are given in Tables 5.E.11 through 5.E.15 and 5.E.16 through 5.E.18, respectively. The stress orientation associated with the stress multipliers for triangular and square hole patterns are shown in Figures 5.E.4 and 5.E.5, respectively. The stress results in paragraph 5.E.7.1 must be transformed to the local x-y-z Cartesian coordinate system before the stress multipliers can be applied.

$$\sigma_{11}=\sigma_{\theta}=\frac{1}{\mu^{*}}\left(K_{x}\sigma_{x}^{*}+K_{y}\sigma_{y}^{*}+K_{xy}\tau_{xy}^{*}\right) \tag{5.E.16}$$

$$\sigma_{12}=\tau_{\theta z}=\frac{1}{\mu^{*}}\left(K_{xz}\tau_{xz}^{*}+K_{yz}\tau_{yz}^{*}\right) \tag{5.E.17}$$

$$\sigma_{13}=0.0 \tag{5.E.18}$$

At the tubeside surface of the plate,

$$\sigma_{22}=-p_{t} \tag{5.E.19}$$

At the shellside surface of the plate,

$$\sigma_{22}=-p_{s} \tag{5.E.20}$$

Between the surfaces of the plate through the thickness,

$$\sigma_{22}=\nu\left(\sigma_{11}-p_{h}\right)+\left(\frac{E}{E_{z}^{*}}\right)\left[\sigma_{z}^{*}-\nu\left(\sigma_{x}^{*}+\sigma_{y}^{*}\right)\right] \tag{5.E.21}$$

and

$$\sigma_{23} = 0.0 \tag{5.E.22}$$

$$\sigma_{33} = -p_h \tag{5.E.23}$$

b) Stress Results At The Rim – the region of the perforated plate outside the diameter D_o is called the plate rim (see Figure 5.E.1). The stress components for the fatigue assessment at the outer most hole shall be computed using the following equations. The stress results in paragraph 5.E.7.1 must be transformed to the local x-y-z Cartesian coordinate system before the stress multipliers can be applied

$$\sigma_{11,rim} = \sigma_{\theta,rim} = K_{\theta,rim}\sigma_{\theta,rim} + K_{r,rim}\sigma_{r,rim} \tag{5.E.24}$$

$$\sigma_{12,rim} = \tau_{\theta z,rim} = K_{rz,rim}\tau_{12,rim} \tag{5.E.25}$$

$$\sigma_{13,rim} = 0.0 \tag{5.E.26}$$

At the tubeside surface of the plate,

$$\sigma_{22,rim} = -p_t \tag{5.E.27}$$

At the shellside surface of the plate,

$$\sigma_{22,rim} = -p_s \tag{5.E.28}$$

Between the surfaces of the plate through the thickness,

$$\sigma_{22,rim} = \nu(\sigma_{11,rim} - p_h) + [\sigma_{z,rim} - \nu(\sigma_{r,rim} + \sigma_{\theta,rim})] \tag{5.E.29}$$

and

$$\sigma_{23,rim} = 0.0 \tag{5.E.30}$$

$$\sigma_{33,rim} = -p_h \tag{5.E.31}$$

The stress multipliers are computed using the information in Tables 5.E.11 through 5.E.18, as applicable, with the following modifications.

$$K_{\theta,rim} = K_x \qquad evaluated\ at\ \mu^* = 1.0 \tag{5.E.32}$$

$$K_{r,rim} = K_y \qquad evaluated\ at\ \mu^* = 1.0 \tag{5.E.33}$$

$$K_{rz,rim} = K_{xz} \qquad evaluated\ at\ \mu^* = 1.0 \tag{5.E.34}$$

c) Thermal Skin Stresses – the temperature gradient through the thickness of a perforated plate can be closely approximated by a step change in the metal temperature near the surface of the plate. Significant thermal stresses develop only in the skin layer of the plate at the surface where the temperature change occurs and the thermal stresses in the remainder of the plate are negligible. The thermal skin stresses at any location on the surface of the equivalent solid plate are given by Equation (5.E.35). The thermal skin stresses should be added to the stresses determined from the numerical analysis

$$\sigma_r^* = \sigma_r^* = K_{skin}\left[\frac{E\alpha\left(T_m - T_s\right)}{\left(1-\nu\right)}\right] \tag{5.E.35}$$

with

$$K_{skin} = \frac{9.43983 - 421.179\mu^* + 6893.05\mu^{*2}}{1 + 4991.39\mu^* + 6032.92\mu^{*2} - 1466.19\mu^{*3}} \tag{5.E.36}$$

d) Stress Results In The Solid Rim And Other Locations – the stress components from the solid rim and other locations that do not contain perforations can be used directly in the fatigue assessment.

5.E.7.3 Stress Range For Fatigue Assessment – The stress range to be used in the fatigue assessment is determined in accordance with paragraph 5.5.

5.E.8 Nomenclature

A	unit cell area
A_0	tube open area
α	coefficient of thermal expansion
β	biaxiality stress factor
$d*$	effective tube hole diameter
d_t	nominal outside diameter of tubes
D_o	equivalent diameter of outer tube limit circle
E	modulus of elasticity for tubesheet material at tubesheet design temperature
E_t	modulus of elasticity for tube material at tube design temperature
E^*	effective Young's modulus for the perforated plate for a triangular hole pattern
E_p^*	effective Young's modulus for the perforated plate for a square hole pattern – pitch direction
E_d^*	effective Young's modulus for the perforated plate for a square hole pattern – diagonal direction
E_z^*	effective Young's modulus for thickness direction loading for perforated plate
E_{11}	coefficient of the elasticity matrix
E_{12}	coefficient of the elasticity matrix
E_{13}	coefficient of the elasticity matrix
E_{21}	coefficient of the elasticity matrix
E_{22}	coefficient of the elasticity matrix
E_{23}	coefficient of the elasticity matrix
E_{31}	coefficient of the elasticity matrix
E_{32}	coefficient of the elasticity matrix
E_{33}	coefficient of the elasticity matrix
E_{44}	coefficient of the elasticity matrix
E_{55}	coefficient of the elasticity matrix
E_{66}	coefficient of the elasticity matrix

F_0 Poisson's ratio factor

F_1 Poisson's ratio factor

G shear modulus for the perforated plate, $G = E/2(1+\nu)$

G^* effective shear modulus for the perforated plate for a triangular hole pattern

G_p^* effective shear modulus for the perforated plate for a square hole pattern – pitch direction

G_d^* effective shear modulus for the perforated plate for a square hole pattern – diagonal direction

G_z^* effective Shear modulus for transverse shear loading for perforated plate.

h tubesheet thickness

K_L stress intensity load factor (see Part 5)

K_{PS} stress multiplier applied to surface stresses to determine local membrane plus bending primary stress averaged across the width of the ligament but not through the thickness.

K_{skin} stress multiplier for the thermal skin stress

K_x Local Stress Multiplier for perforated material

K_y Local Stress Multiplier for perforated material

K_{xy} Local Stress Multiplier for perforated material

K_{xz} Local Stress Multiplier for perforated material

K_{yz} Local Stress Multiplier for perforated material

$K_{r,rim}$ Local Stress Multiplier for solid rim plate

$K_{rz,rim}$ Local Stress Multiplier for solid rim plate

$K_{\theta,rim}$ Local Stress Multiplier for solid rim plate

l_{tx} expanded length of tube in tubesheet $(0.0 \le l_{tx} \le 1.0)$. An expanded tube-to-tubesheet joint is produced by applying pressure inside the tube such that contact is established between the tube and tubesheet. In selecting an appropriate value of expanded length, the designer shall consider the degree of initial expansion, differences in thermal expansion, or other factors that could result in loosening of the tubes within the tubesheet (see Figure 4.18.2.Sketch (b)).

μ ligament efficiency

μ^* effective ligament efficiency

ν Poisson's ratio for the perforated plate

ν^* effective Poisson's ratio for the perforated plate for a triangular hole pattern

ν_p^* effective Poisson's ratio for the perforated plate for a square hole pattern – pitch direction

ν_d^* effective Poisson's ratio for the perforated plate for a square hole pattern – diagonal direction

P_b primary bending stress intensity.

P_m general primary membrane stress intensity.

P_L local primary membrane stress intensity.

p tube pitch

p^* effective tube pitch

p_s shellside pressure acting on the surface of the plate

p_t tubeside pressure acting on the surface of the plate

p_h pressure acting in the hole of the plate

p_1 tubeside pressure acting on the surface of the plate

p_2 shellside pressure acting on the surface of the plate

p_3 shellside pressure acting on the surface of the rim

p_4 pressure acting on the shell

p_5 tubeside pressure acting on the surface of the rim

Q secondary stress intensity.

r_o radius to outermost tube hole center

ρ tube expansion depth ratio; $0.0 \le \rho \le 1.0$

S_m allowable stress intensity for the perforated plate material at the design temperature

S_{PS} allowable limit on the primary plus secondary stress range.

S_{tm} allowable stress intensity for the tube material at the design temperature

ε_r^* calculated radial strain from the stress analysis of a equivalent perforated plate

ε_θ^* calculated circumferential strain from the stress analysis of a equivalent perforated plate

ε_z^* calculated axial strain from the stress analysis of a equivalent perforated plate

$\gamma_{r\theta}^*$ calculated shear strain from the stress analysis of a equivalent perforated plate

γ_{rz}^* calculated shear strain from the stress analysis of a equivalent perforated plate

$\gamma_{z\theta}^*$ calculated shear strain from the stress analysis of a equivalent perforated plate

σ_{11} normal stress in the 1-direction

σ_{12} shear stress in the 1-direction on the 2-plane

σ_{13} shear stress in the 1-direction on the 3-plane

σ_{22} normal stress in the 2-direction

σ_{23} shear stress in the 2-direction on the 3-plane

σ_{33} stress in the 3-direction

$\sigma_{11,rim}$ normal stress in the 1-direction at the rim

$\sigma_{12,rim}$ shear stress in the 1-direction on the 2-plane at the rim

$\sigma_{13,rim}$ shear stress in the 1-direction on the 3-plane at the rim

$\sigma_{22,rim}$ normal stress in the 2-direction at the rim

$\sigma_{23,rim}$ shear stress in the 2-direction on the 3-plane at the rim

$\sigma_{33,rim}$ stress in the 3-direction at the rim

σ_r^* normal radial stress calculated from the stress analysis of a equivalent perforated plate

σ_θ^* normal circumferential stress calculated from the stress analysis of a equivalent perforated plate

σ_z^* normal axial stress calculated from the stress analysis of a equivalent perforated plate

$\tau_{r\theta}^*$ normal shear stress calculated from the stress analysis of a equivalent perforated plate

τ_{rz}^{*}	normal shear stress calculated from the stress analysis of a equivalent perforated plate
$\tau_{z\theta}^{*}$	normal shear stress calculated from the stress analysis of a equivalent perforated plate
$\left(\sigma_{r}^{*}\right)_{a}$	normal radial stress from the stress analysis of the equivalent plate
$\left(\sigma_{\theta}^{*}\right)_{a}$	normal circumferential stress from the stress analysis of the equivalent plate
$\left(\sigma_{r}^{*}\right)_{d}$	normal radial stress computed from the elasticity matrix for the square hole pattern in the diagonal direction
$\left(\sigma_{\theta}^{*}\right)_{d}$	normal circumferential stress computed from the elasticity matrix for the square hole pattern in the diagonal direction
$\left(\sigma_{r}^{*}\right)_{p}$	normal radial stress computed from the elasticity matrix for the square hole pattern in the pitch direction
$\left(\sigma_{\theta}^{*}\right)_{p}$	normal circumferential stress computed from the elasticity matrix for the square hole pattern in the pitch direction
σ_{r}^{*}	normal radial stress at the perforated plate to solid ring interface calculated from the stress analysis
σ_{θ}^{*}	normal circumferential stress at the perforated plate to solid ring interface calculated from the stress analysis
τ_{rz}^{*}	shear stress at the perforated plate to solid ring interface calculated from the stress analysis
t_{t}	nominal tube wall thickness
T_{m}	mean temperature averaged through the thickness of the plate
T_{s}	temperature at the surface of the plate
U_{L}	largest center-to-center distance between adjacent tube rows, but not to exceed $4p$
θ	orientation for the stress calculation in the referenced to the local Cartesian coordinate system (see Figures 5.E.4 and 5.E.5)

5.E.9 Tables

Table 5.E.1 – Values of E^* For Perforated Tubesheets With An Equilateral Triangular Pattern

Coefficients	h/p			
	0.1	0.25	0.5	2.0
A_0	1.496410E-05	6.352630E-06	-1.333850E-03	7.285910E-04
A_1	6.620641E+01	1.151132E+03	-2.556960E+00	-7.005325E-01
A_2	4.301938E+00	1.275654E+01	6.777985E-01	1.151695E-01
A_3	-3.393999E+01	-1.643137E+03	3.994838E+00	2.440697E+00
A_4	7.672133E+01	1.202564E+03	1.550319E-01	4.439786E+00
A_5	1.497235E+00	1.257743E+03	-4.396305E+00	-5.840415E+00
A_6	-3.012851E+01	-9.998214E+02	-2.431697E+00	-8.312674E+00
A_7	1.613346E+01	-5.511521E+02	2.177133E+00	4.123150E+00
A_8	0.0	0.0	1.819010E+00	4.780299E+00

Note:

1. $E^* = E\left[\frac{A_0 + A_2\mu^* + A_4\mu^{*2} + A_6\mu^{*3} + A_8\mu^{*4}}{1 + A_1\mu^* + A_3\mu^{*2} + A_5\mu^{*3} + A_7\mu^{*4}}\right]$
2. These coefficients are valid for $0 \le \mu^* \le 1.0$, data for $\mu^* = 2.0$ is provided for information only.
3. If $h/p < 0.1$, then use $h/p = 0.1$.
4. If $h/p > 2.0$, then use $h/p = 2.0$.

Table 5.E.2 – Values of ν^* For Perforated Tubesheets With An Equilateral Triangular Pattern

Coefficients	h/p					
	0.1	0.15	0.25	0.5	1.0	2.0
B_0	1.722338E-02	-7.248304E-01	3.824487E+02	2.860400E+00	1.483591E+00	9.823512E-01
B_1	-3.203150E+01	-3.180156E+01	1.539472E+04	5.679314E+01	1.504975E+01	9.655558E-01
B_2	1.765880E+00	1.599207E+01	2.264094E+03	4.717135E-01	-1.976814E+00	-2.811381E+00
B_3	2.321240E+02	2.859222E+02	-4.175812E+04	-2.166572E+02	-6.193608E+01	4.633821E+00
B_4	-2.593198E+01	-9.517465E+01	-1.461539E+04	-3.787845E+01	-7.869233E+00	5.917858E+00
B_5	-2.171148E+02	-1.388896E+02	7.152697E+04	3.226507E+02	1.094078E+02	7.097756E+00
B_6	9.357422E+01	2.547347E+02	2.981678E+04	8.368412E+01	2.733155E+01	0.0
B_7	1.107294E+02	7.201617E+01	1.387188E+04	0.0	0.0	0.0
B_8	-4.095396E+01	-1.182527E+02	0.0	0.0	0.0	0.0

Note:

1. $\nu^* = \dfrac{B_0 + B_2\mu^* + B_4\mu^{*2} + B_6\mu^{*3} + B_8\mu^{*4}}{1 + B_1\mu^* + B_3\mu^{*2} + B_5\mu^{*3} + B_7\mu^{*4}}$
2. These coefficients are valid for $0 \le \mu^* \le 1.0$ except if $h/p = 0.1$, then $0.184 \le \mu^* \le 1.0$, data for $\mu^* = 2.0$ is provided for information only.
3. If $h/p < 0.1$, then use $h/p = 0.1$.
4. If $h/p > 2.0$, then use $h/p = 2.0$.

Table 5.E.3 – Values of E^* For Perforated Tubesheets With A Square Pattern

Coefficients	h/p			
	0.1	0.25	0.5	2.0
A_0	-4.085600E-05	-4.828500E-04	8.925870E-05	1.589308E-03
A_1	5.231431E+02	8.372334E+00	1.783577E+01	1.814855E+01
A_2	-6.376136E+00	2.947067E+00	2.896931E+00	2.693793E+00
A_3	7.857623E+03	-9.963856E+00	-1.945644E+01	-2.374421E+01
A_4	2.284635E+03	3.710254E+00	1.377083E+01	9.002308E+00
A_5	-7.366960E+03	7.250874E+00	8.840783E+00	1.627106E+01
A_6	4.844156E+03	0.0	-8.457560E+00	0.0
A_7	6.092597E+03	0.0	0.0	0.0

Note:

1. $$E^* = E\left[\frac{A_0 + A_2\mu^* + A_4\mu^{*2} + A_6\mu^{*3}}{1 + A_1\mu^* + A_3\mu^{*2} + A_5\mu^{*3} + A_7\mu^{*4}}\right]$$
2. These coefficients are valid for $0 \le \mu^* \le 1.0$, data for $\mu^* = 2.0$ is provided for information only.
3. If $h/p < 0.1$, then use $h/p = 0.1$.
4. If $h/p > 2.0$, then use $h/p = 2.0$.

Table 5.E.4 – Values of ν^* For Perforated Tubesheets With A Square Pattern

Coefficients	h/p					
	0.1	0.15	0.25	0.5	1.0	2.0
B_0	-7.288894E-02	2.089015E+00	1.397869E+03	4.915542E-01	3.475156E-01	3.337833E-01
B_1	-1.127111E+01	1.199162E+02	8.678686E+04	2.202591E+01	-1.094083E+00	2.923681E+00
B_2	1.325618E+00	-1.535964E+01	5.003924E+03	4.392312E+00	-5.737278E-01	1.132010E+00
B_3	3.949776E+01	-7.543170E+02	-1.857192E+05	-5.969066E+01	4.985966E+00	-1.209380E+01
B_4	-8.530838E+00	6.129051E+01	-2.174219E+04	-1.407781E+01	1.875825E+00	-4.921427E+00
B_5	-2.359313E+01	1.921504E+03	2.363003E+05	6.022030E+01	6.108286E-01	1.656896E+01
B_6	2.130422E+01	-1.346693E+02	5.674145E+04	1.627461E+01	0.0	6.722375E+00
B_7	5.916208E+00	-1.921418E+03	0.0	0.0	0.0	2.496675E+00
B_8	-1.056089E+01	1.808025E+02	0.0	0.0	0.0	0.0
B_9	0.0	9.465777E+02	0.0	0.0	0.0	0.0

Note:

1. $\nu^* = \dfrac{B_0 + B_2\mu^* + B_4\mu^{*2} + B_6\mu^{*3} + B_8\mu^{*4}}{1 + B_1\mu^* + B_3\mu^{*2} + B_5\mu^{*3} + B_7\mu^{*4} + B_9\mu^{*5}}$
2. These coefficients are valid for $0 \le \mu^* \le 1.0$, data for $\mu^* = 2.0$ is provided for information only.
3. If $h/p < 0.1$, then use $h/p = 0.1$.
4. If $h/p > 2.0$, then use $h/p = 2.0$.

Table 5.E.5 – Effective Elastic Modulus, Poisson's Ratio, And Shear Modulus For A Perforated Plate With A Triangular Hole Pattern

Plane Stress

$$E^* = E\begin{bmatrix} -0.000710796 + 0.138017\mu^* + 4.22004\mu^{*2} - \\ 7.55885\mu^{*3} + 7.08667\mu^{*4} - 2.88588\mu^{*5} \end{bmatrix}$$

$$\nu^* = 0.999804 - 3.94540\mu^* + 8.47377\mu^{*2} - 8.02434\mu^{*3} + 2.79724\mu^{*4}$$

$$G^* = \frac{E^*}{2\left(1+\nu^*\right)}$$

Generalized Plane Strain

$$\overline{E}^* = \left[\frac{\left(1-\nu^2\right)}{E^*} + \frac{\nu^2}{E_z^*}\right]^{-1}$$

$$\overline{\nu}^* = \overline{E}^*\left[\frac{\left(1-\nu^2\right)\nu^*}{E^*} + \frac{\nu^3}{E} + \nu^2\left(\frac{1}{E} + \frac{1}{E_z^*}\right)\right]$$

$$\overline{G}^* = G^*$$

Plane Stress and Generalized Plane Strain

$$E_z^* = E\left[1 - \frac{\pi\left(1-\mu^*\right)^2}{2\sqrt{3}}\right]$$

$$G_z^* = G\left[\frac{0.000209547 + 3.00960\mu^* + 32.7845\mu^{*2} - 16.57921\mu^{*3}}{1 + 34.7831\mu^* - 33.7539\mu^{*2} + 17.1860\mu^{*3}}\right]$$

Notes: These coefficients are only valid for $0.1 \le \mu^* \le 1.0$.

Table 5.E.6 – Effective Elastic Modulus, Poisson's Ratio, And Shear Modulus For A Perforated Plate With A Square Hole Pattern – Pitch Direction

Plane Stress

$$E_p^* = E\begin{bmatrix} 0.00381121 + 2.98282\mu^* - 17.5077\mu^{*2} + 79.3742\mu^{*3} - \\ 193.933\mu^{*4} + 258.876\mu^{*5} - 177.201\mu^{*6} + 48.4060\mu^{*7} \end{bmatrix}$$

$$\nu_p^* = \frac{0.000479230 + 2.22119\mu^* + 6.92863\mu^{*2} - 7.35650\mu^{*3}}{1 + 22.8486\mu^* - 37.2914\mu^{*2} + 46.4082\mu^{*3} - 26.9865\mu^{*4}}$$

$$G_p^* = \frac{E_d^*}{2\left(1+\nu_d^*\right)}$$

Generalized Plane Strain

$$\overline{E}_p^* = \left[\frac{\left(1-\nu^2\right)}{E_p^*} + \frac{\nu^2}{E_z^*}\right]^{-1}$$

$$\overline{\nu}^* = \overline{E}_p^*\left[\frac{\left(1-\nu^2\right)\nu_p^*}{E_p^*} + \frac{\nu^3}{E} + \nu^2\left(\frac{1}{E} + \frac{1}{E_z^*}\right)\right]$$

$$\overline{G}_p^* = G_p^*$$

Plane Stress and Generalized Plane Strain

$$E_z^* = E\left[1 - \frac{\pi\left(1-\mu^*\right)^2}{4}\right]$$

$$G_z^* = G\left[\frac{0.0017 + 4.42892\mu^* + 23.3350\mu^{*2} - 12.8075\mu^{*3}}{1 + 25.7230\mu^* - 22.3355\mu^{*2} + 10.5712\mu^{*3}}\right]$$

Notes:

1. These coefficients are only valid for $0.1 \le \mu^* \le 1.0$.
2. The constants E_d^* and ν_d^* can be determined using the equations in Table 5.E.7.

Table 5.E.7 – Effective Elastic Modulus, Poisson's Ratio, And Shear Modulus For A Perforated Plate With A Square Hole Pattern – Diagonal Direction

<u>**Plane Stress**</u>

$$E_d^* = E\left[\frac{-0.000399053 + 0.165345\mu^* + 2.02690\mu^{*2} - 1.15319\mu^{*3}}{1 - 0.493831\mu^* + 0.354744\mu^{*2} + 0.177698\mu^{*3}}\right]$$

$$\nu_d^* = \frac{0.997998 - 2.27993\mu^* + 1.96210\mu^{*2}}{1 - 0.250614\mu^* - 1.330\mu^{*2} + 2.85019\mu^{*3}}$$

$$G_d^* = \frac{E_p^*}{2\left(1 + \nu_p^*\right)}$$

<u>**Generalized Plane Strain**</u>

$$\overline{E}_d^* = \left[\frac{\left(1 - \nu^2\right)}{E_d^*} + \frac{\nu^2}{E_z^*}\right]^{-1}$$

$$\overline{\nu}^* = \overline{E}_d^*\left[\frac{\left(1 - \nu^2\right)\nu_d^*}{E_d^*} + \frac{\nu^3}{E} + \nu^2\left(\frac{1}{E} + \frac{1}{E_z^*}\right)\right]$$

$$\overline{G}_d^* = G_d^*$$

<u>**Plane Stress and Generalized Plane Strain**</u>

$$E_z^* = E\left[1 - \frac{\pi\left(1 - \mu^*\right)^2}{4}\right]$$

$$G_z^* = G\left[\frac{0.0017 + 4.42892\mu^* + 23.3350\mu^{*2} - 12.8075\mu^{*3}}{1 + 25.7230\mu^* - 22.3355\mu^{*2} + 10.5712\mu^{*3}}\right]$$

Notes:

1. These coefficients are only valid for $0.1 \le \mu^* \le 1.0$.
2. The constants E_p^* and ν_p^* can be determined using the equations in Table 5.E.6.

Table 5.E.8 – Orthotropic Effective Elasticity Matrix For A Perforated Plate With An Equilateral Triangular Hole Pattern

$$\begin{bmatrix} \sigma_r^* \\ \sigma_\theta^* \\ \sigma_z^* \\ \tau_{\theta z}^* \\ \tau_{rz}^* \\ \tau_{r\theta}^* \end{bmatrix} = \begin{bmatrix} E_{11} & E_{12} & E_{13} & 0 & 0 & 0 \\ E_{21} & E_{22} & E_{23} & 0 & 0 & 0 \\ E_{31} & E_{32} & E_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & E_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & E_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & E_{66} \end{bmatrix} \cdot \begin{bmatrix} \varepsilon_r^* \\ \varepsilon_\theta^* \\ \varepsilon_z^* \\ \gamma_{\theta z}^* \\ \gamma_{rz}^* \\ \gamma_{r\theta}^* \end{bmatrix}$$

$$E_{11} = E_{22} = \frac{E^*}{H \cdot (1+\nu^*)}\left[1 - \left(\frac{E^*}{E_z^*}\right)\nu^2\right]$$

$$E_{12} = E_{21} = \frac{E^*}{H \cdot (1+\nu^*)}\left[\nu^* - \left(\frac{E^*}{E_z^*}\right)\nu^2\right]$$

$$E_{13} = E_{31} = E_{32} = E_{23} = \frac{E^*\nu}{H}$$

$$E_{33} = \frac{E_z^*(1-\nu^*)}{H}$$

$$E_{44} = E_{55} = G_z^*$$

$$E_{66} = \frac{E^*}{2(1+\nu^*)}$$

where:

$$H = 1 - \nu^* - 2\left(\frac{E^*}{E_z^*}\right)\nu^2$$

Table 5.E.9 – Orthotropic Effective Elasticity Matrix For A Perforated Plate With A Square Hole Pattern

$$\begin{bmatrix} \sigma_r^* \\ \sigma_\theta^* \\ \sigma_z^* \\ \tau_{\theta z}^* \\ \tau_{rz}^* \\ \tau_{r\theta}^* \end{bmatrix} = \begin{bmatrix} E_{11} & E_{12} & E_{13} & 0 & 0 & 0 \\ E_{21} & E_{22} & E_{23} & 0 & 0 & 0 \\ E_{31} & E_{32} & E_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & E_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & E_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & E_{66} \end{bmatrix} \cdot \begin{bmatrix} \varepsilon_r^* \\ \varepsilon_\theta^* \\ \varepsilon_z^* \\ \gamma_{\theta z}^* \\ \gamma_{rz}^* \\ \gamma_{r\theta}^* \end{bmatrix}$$

$$E_{11} = E_{22} = \frac{E_p^*}{H \cdot (1+\nu_p^*)}\left[1 - \left(\frac{E_p^*}{E_z^*}\right)\nu^2\right] + \frac{1}{2}\left[G_p^* - \frac{E_p^*}{2(1-\nu_p^*)}\right]$$

$$E_{12} = E_{21} = \frac{E_p^*}{H \cdot (1+\nu_p^*)}\left[\nu_p^* + \nu^2\left(\frac{E_p^*}{E_z^*}\right)\right] - \frac{1}{2}\left[G_p^* - \frac{E_p^*}{2(1-\nu_p^*)}\right]$$

$$E_{13} = E_{31} = E_{32} = E_{23} = \frac{E_p^* \nu}{H}$$

$$E_{33} = \frac{E_z^*(1-\nu_p^*)}{H}$$

$$E_{44} = E_{55} = G_z^*$$

$$E_{66} = G_p^* = \frac{E_d^*}{2(1+\nu_d^*)}$$

where:

$$H = 1 - \nu_p^* - 2\left(\frac{E_p^*}{E_z^*}\right)\nu^2$$

Note: The effective constants in diagonal directions ν_d^*, E_d^*, G_d^* can be equivalently used instead of pitch direction constants ν_p^*, E_p^*, G_p^* in the elasticity matrix shown above.

Table 5.E.10 – Equations For Determining Stress Components Based On The Results From An Equivalent Plate Analysis For An Equilateral Rectangular Hole Pattern

$$\begin{bmatrix} \left(\sigma_r^*\right)_a \\ \left(\sigma_\theta^*\right)_a \\ \left(\sigma_z^*\right)_a \\ \left(\tau_{\theta z}^*\right)_a \\ \left(\tau_{rz}^*\right)_a \\ \left(\tau_{r\theta}^*\right)_a \end{bmatrix} = \begin{bmatrix} E_{11} & E_{12} & E_{13} & 0 & 0 & 0 \\ E_{22} & E_{22} & E_{23} & 0 & 0 & 0 \\ E_{33} & E_{33} & E_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & E_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & E_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & E_{66} \end{bmatrix} \cdot \begin{bmatrix} \varepsilon_r^* \\ \varepsilon_\theta^* \\ \varepsilon_z^* \\ \gamma_{\theta z}^* \\ \gamma_{rz}^* \\ \gamma_{r\theta}^* \end{bmatrix}$$

$$E_{11} = E_{22} = \frac{E_p^*}{H \cdot (1+\nu_p^*)}\left[1-\left(\frac{E_p^*}{E_z^*}\right)\nu^2\right] + 4\sin^2\theta\cos^2\theta\left[G_p^* - \frac{E_p^*}{2(1+\nu_p^*)}\right]$$

$$E_{12} = E_{21} = \frac{E_p^*}{H \cdot (1+\nu_p^*)}\left[\nu_p^* + \nu^2\left(\frac{E_p^*}{E_z^*}\right)\right] + 4\sin^2\theta\cos^2\theta\left[G_p^* - \frac{E_p^*}{2(1+\nu_p^*)}\right]$$

$$E_{13} = E_{31} = E_{32} = E_{23} = \frac{E_p^*\nu}{H}$$

$$E_{14} = E_{41} = -E_{24} = -E_{42} = 2\left[G_P^* - \frac{E_p^*}{2\left(1+\nu_p^*\right)}\right]\sin\theta\cos\theta\left(\cos^2\theta - \sin^2\theta\right)$$

$$E_{33} = \frac{E_z^*(1-\nu_p^*)}{H}$$

$$E_{44} = E_{55} = G_z^*$$

$$E_{66} = G_p^* - 4\sin^2\theta\cos^2\theta\left[G_p^* - \frac{E_p^*}{2(1+\nu_p^*)}\right]$$

Note:

1. The constants $E_p^*, \nu_P^*, E_z^*, G_z^*, H, \nu$ are evaluated from Table 5.E.5 through 5.E.9.
2. The angle θ defines the orientation of the radial direction in the equivalent solid plate with respect to the pitch direction of the square hole pattern.

Table 5.E.11 – Stress Factor K_x Coefficients – Triangular Hole Pattern

μ^*	C_1	C_2	C_3	C_4	C_5
0.05	-2.48810E-03	0.00000E+00	2.29959E-04	0.00000E+00	-8.60430E-07
0.1	-4.15591E-03	0.00000E+00	3.99110E-04	0.00000E+00	-1.66750E-06
0.15	9.71786E-03	0.00000E+00	2.27507E-04	0.00000E+00	-2.02240E-06
0.2	2.44173E-02	-2.05029E-03	-1.50600E-04	2.92198E-06	2.80179E-08
0.25	4.97018E-03	-4.07320E-04	-4.68940E-04	1.34485E-05	-7.98040E-08
0.3	-4.38107E-02	1.63067E-03	-6.65130E-04	1.92456E-05	-1.14750E-07
0.333	-9.22413E-02	3.29146E-03	-6.93000E-04	2.32530E-05	-2.12050E-07
0.4	-2.09998E-01	6.59092E-03	-5.32060E-04	1.63467E-05	-1.04550E-07
0.5	-4.01999E-01	-3.94910E-04	2.45399E-04	-1.44260E-05	1.19709E-06
0.6	-5.75989E-01	-1.66130E-04	5.74246E-04	-4.90600E-06	3.73271E-07
0.7	-7.14696E-01	-8.35510E-05	8.53270E-04	-7.23000E-06	5.22499E-07
0.8	-8.19200E-01	-8.61900E-04	1.26476E-03	-3.60460E-05	2.33690E-06
0.9	-9.07202E-01	-3.10980E-04	1.16915E-03	-6.47650E-06	2.87057E-07
1	-1.00000E+00	-6.01550E-04	1.40134E-03	-2.02820E-05	1.00488E-06
μ^*	C_6	C_7	C_8	C_9	C_{10}
0.05	0.00000E+00	3.79570E-10	0.00000E+00	-5.00380E-14	0.00000E+00
0.1	0.00000E+00	7.52024E-10	0.00000E+00	-9.80750E-14	0.00000E+00
0.15	0.00000E+00	9.96594E-10	0.00000E+00	-1.38990E-13	0.00000E+00
0.2	-3.48070E-10	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.25	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.3	-6.26280E-11	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.333	4.79506E-10	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.4	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.5	-3.91660E-08	8.96544E-10	-1.42550E-11	1.38091E-13	-7.22820E-16
0.6	-1.21420E-08	2.55297E-10	-3.89990E-12	3.65760E-14	-1.81670E-16
0.7	-2.46600E-08	6.39236E-10	-1.03660E-11	1.01715E-13	-5.48690E-16
0.8	-9.59670E-08	2.34704E-09	-3.58780E-11	3.34855E-13	-1.74180E-15
0.9	-1.57740E-08	4.06586E-10	-6.50940E-12	6.42221E-14	-3.52630E-16
1	-3.55490E-08	6.67355E-10	-7.22950E-12	4.25909E-14	-1.05160E-16
μ^*	C_{11}	C_{12}	C_{13}	C_{14}	C_{15}
0.05	2.03746E-18	0.00000E+00	-1.02414E-03	0.00000E+00	6.61817E-07
0.1	4.37065E-18	0.00000E+00	-6.92440E-04	0.00000E+00	6.50223E-07
0.15	7.88278E-18	0.00000E+00	5.49459E-07	0.00000E+00	2.94340E-07
0.2	0.00000E+00	-7.40767E-02	2.51869E-03	-4.42910E-05	3.93678E-07
0.25	0.00000E+00	-5.46478E-02	1.59424E-03	-2.61460E-05	2.33088E-07
0.3	0.00000E+00	-3.98100E-02	1.04897E-03	-1.75510E-05	1.75088E-07
0.333	0.00000E+00	-3.57930E-02	8.10224E-04	-1.19240E-05	1.11006E-07
0.4	0.00000E+00	-3.13608E-02	7.61981E-04	-1.15130E-05	1.06015E-07
0.5	1.57382E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.6	3.65531E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.7	1.24975E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.8	3.86817E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.9	8.21600E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
1	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00

Table 5.E.11 – Stress Factor K_x Coefficients – Triangular Hole Pattern

μ^*	C_{16}	C_{17}	C_{18}	C_{19}	
0.05	0.00000E+00	-2.21770E-10	3.53954E-14	-1.97610E-18	
0.1	0.00000E+00	-2.34390E-10	3.92678E-14	-2.04780E-18	
0.15	0.00000E+00	-8.74970E-11	1.41889E-14	-6.43590E-20	
0.2	-1.36300E-09	0.00000E+00	0.00000E+00	0.00000E+00	
0.25	-8.31760E-10	0.00000E+00	0.00000E+00	0.00000E+00	
0.3	-7.17880E-10	0.00000E+00	0.00000E+00	0.00000E+00	
0.333	-4.62140E-10	0.00000E+00	0.00000E+00	0.00000E+00	
0.4	-4.07500E-10	0.00000E+00	0.00000E+00	0.00000E+00	
0.5	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.6	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.7	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.8	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.9	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
1	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	

Notes:

1. $$K_x = \frac{C_1 + C_2\theta + C_3\theta^2 + C_4\theta^3 + C_5\theta^4 + C_6\theta^5 + C_7\theta^6 + C_8\theta^7 + C_9\theta^8 + C_{10}\theta^9 + C_{11}\theta^{10}}{1 + C_{12}\theta + C_{13}\theta^2 + C_{14}\theta^3 + C_{15}\theta^4 + C_{16}\theta^5 + C_{17}\theta^6 + C_{18}\theta^8 + C_{19}\theta^{10}}$$
2. See Figure 5.E.4 for definition of K_x and θ (θ is in Degrees, $0 \le \theta \le 90$)
3. Valid range is $0.1 \le \mu^* \le 1.0$, data for $\mu^* = 0.05$ is provided for information only.

Table 5.E.12 – Stress Factor K_y Coefficients – Triangular Hole Pattern

μ^*	C_1	C_2	C_3	C_4	C_5
0.05	1.03339E+00	0.00000E+00	-2.84109E-03	0.00000E+00	4.19314E-06
0.1	1.07512E+00	0.00000E+00	-2.27215E-03	0.00000E+00	3.65504E-06
0.15	1.10022E+00	0.00000E+00	-1.22107E-03	0.00000E+00	2.78469E-06
0.2	1.11688E+00	-8.46125E-02	2.79641E-03	-2.96440E-05	-2.88860E-08
0.25	1.16107E+00	-5.62139E-02	1.52480E-03	-5.12960E-06	-2.84200E-07
0.3	1.23266E+00	-7.01751E-02	2.10035E-03	-2.92840E-05	1.38335E-07
0.333	1.29302E+00	0.00000E+00	9.21649E-04	0.00000E+00	1.47528E-07
0.4	1.44919E+00	-5.25533E-02	1.13847E-03	-1.51710E-05	7.22991E-08
0.5	1.71400E+00	-2.27180E-05	-2.32310E-04	5.53814E-07	-2.02130E-07
0.6	1.98420E+00	2.88144E-04	-6.45880E-04	9.50408E-06	-5.54150E-07
0.7	2.24001E+00	-2.21010E-04	-7.35030E-04	-1.06590E-05	7.63943E-07
0.8	2.47998E+00	-9.56630E-06	-9.88900E-04	2.31615E-07	1.03079E-07
0.9	2.72430E+00	-6.11640E-05	-1.11908E-03	3.82662E-06	-2.21980E-07
1	3.00000E+00	6.01553E-04	-1.40134E-03	2.02817E-05	-1.00490E-06

μ^*	C_6	C_7	C_8	C_9	C_{10}
0.05	0.00000E+00	-1.64540E-09	0.00000E+00	2.29250E-13	0.00000E+00
0.1	0.00000E+00	-1.39530E-09	0.00000E+00	1.83217E-13	0.00000E+00
0.15	0.00000E+00	-1.06820E-09	0.00000E+00	1.44899E-13	0.00000E+00
0.2	9.69984E-10	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.25	2.10463E-09	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.3	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.333	0.00000E+00	-1.56830E-10	0.00000E+00	1.06849E-14	0.00000E+00
0.4	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.5	4.62233E-09	-1.14270E-10	2.12833E-12	-1.80180E-14	5.56084E-17
0.6	1.65435E-08	-2.97480E-10	3.50182E-12	-2.37960E-14	7.59107E-17
0.7	-2.78200E-08	6.69142E-10	-9.94040E-12	8.96030E-14	-4.52210E-16
0.8	-1.72350E-09	7.31083E-11	-1.64730E-12	1.93694E-14	-1.18190E-16
0.9	1.52399E-08	-4.17110E-10	6.84700E-12	-6.81690E-14	3.74951E-16
1	3.55493E-08	-6.67360E-10	7.22950E-12	-4.25910E-14	1.05163E-16

μ^*	C_{11}	C_{12}	C_{13}	C_{14}	C_{15}
0.05	2.03746E-18	0.00000E+00	-1.02414E-03	0.00000E+00	6.61817E-07
0.05	-1.10020E-17	0.00000E+00	-1.30100E-04	0.00000E+00	1.62821E-07
0.1	-8.94190E-18	0.00000E+00	-4.43300E-04	0.00000E+00	5.47099E-07
0.15	-1.02870E-17	0.00000E+00	9.13766E-05	0.00000E+00	1.17503E-07
0.2	0.00000E+00	-7.50744E-02	2.84658E-03	-5.41970E-05	5.06334E-07
0.25	0.00000E+00	-4.84213E-02	1.54856E-03	-2.82640E-05	2.81005E-07
0.3	0.00000E+00	-5.72395E-02	1.69354E-03	-2.71490E-05	2.33083E-07
0.333	0.00000E+00	0.00000E+00	5.88781E-04	0.00000E+00	-2.31450E-08
0.4	0.00000E+00	-3.63267E-02	7.41010E-04	-8.51480E-06	5.71068E-08
0.5	-7.05120E-21	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.6	-6.43060E-20	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.7	9.79922E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.8	2.94832E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.9	-8.72370E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
1.0	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00

Table 5.E.12 – Stress Factor K_y Coefficients – Triangular Hole Pattern

μ^*	C_{16}	C_{17}	C_{18}	C_{19}	
0.05	0.00000E+00	-1.12070E-10	2.48827E-14	-1.26920E-18	
0.1	0.00000E+00	-2.01280E-10	3.33735E-14	-1.26370E-18	
0.15	0.00000E+00	2.78911E-11	-1.41260E-14	3.28174E-18	
0.2	-1.68980E-09	0.00000E+00	0.00000E+00	0.00000E+00	
0.25	-1.01320E-09	0.00000E+00	0.00000E+00	0.00000E+00	
0.3	-8.14480E-10	0.00000E+00	0.00000E+00	0.00000E+00	
0.333	0.00000E+00	2.63945E-11	-7.08690E-16	0.00000E+00	
0.4	-1.57550E-10	0.00000E+00	0.00000E+00	0.00000E+00	
0.5	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.6	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.7	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.8	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.9	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
1	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	

Notes:

1. $$K_y = \frac{C_1 + C_2\theta + C_3\theta^2 + C_4\theta^3 + C_5\theta^4 + C_6\theta^5 + C_7\theta^6 + C_8\theta^7 + C_9\theta^8 + C_{10}\theta^9 + C_{11}\theta^{10}}{1 + C_{12}\theta + C_{13}\theta^2 + C_{14}\theta^3 + C_{15}\theta^4 + C_{16}\theta^5 + C_{17}\theta^6 + C_{18}\theta^8 + C_{19}\theta^{10}}$$
2. See Figure 5.E.4 for definition of K_y and θ (θ is in Degrees, $0 \le \theta \le 90$)
3. Valid range is $0.1 \le \mu^* \le 1.0$, data for $\mu^* = 0.05$ is provided for information only.

Table 5.E.13 – Stress Factor K_{xy} Coefficients – Triangular Hole Pattern

μ^*	C_1	C_2	C_3	C_4	C_5
0.05	-7.39914E-03	-1.08535E+00	8.22248E-02	-2.38787E-03	3.01315E-05
0.1	1.01945E-03	-5.69877E-01	4.14104E-02	-1.17352E-03	1.45770E-05
0.15	6.42492E-04	-3.76414E-01	2.49867E-02	-6.82750E-04	8.40611E-06
0.2	2.06985E-04	-2.82692E-01	3.19868E-03	-1.11430E-04	3.35540E-05
0.25	8.93706E-05	-2.26878E-01	1.27974E-03	2.90909E-05	1.42069E-05
0.3	4.18418E-05	-1.91445E-01	6.07889E-04	4.80215E-05	7.84724E-06
0.333	-6.40560E-06	-1.73543E-01	5.82276E-03	-6.51600E-05	2.43193E-07
0.4	3.27439E-06	-1.49979E-01	3.96703E-03	-2.72410E-05	-6.26590E-08
0.5	9.56664E-07	-1.27747E-01	4.80727E-05	4.22320E-05	5.50887E-07
0.6	2.03447E-06	-1.15912E-01	-9.13650E-06	3.52216E-05	-3.00780E-08
0.7	-5.55990E-06	-1.13285E-01	8.09445E-05	1.53690E-05	8.34663E-07
0.8	1.24527E-05	-1.17560E-01	3.80773E-05	2.14282E-05	1.42560E-07
0.9	5.89104E-06	-1.26273E-01	-1.97790E-04	4.88674E-05	-1.39820E-06
1	-5.06060E-06	-1.39930E-01	7.41931E-05	2.11002E-05	3.89961E-07

μ^*	C_6	C_7	C_8	C_9	C_{10}
0.05	-1.36240E-07	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.1	-6.52070E-08	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.15	-3.76500E-08	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.2	-1.81580E-06	4.43342E-08	-5.67960E-10	3.77794E-12	-1.10280E-14
0.25	-8.15470E-07	1.95441E-08	-2.39950E-10	1.46392E-12	-3.30050E-15
0.3	-4.62740E-07	1.13561E-08	-1.49140E-10	1.05550E-12	-3.56840E-15
0.333	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.4	9.03506E-10	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.5	-3.69530E-08	7.98203E-10	-1.17410E-11	1.14039E-13	-6.19170E-16
0.6	-4.53810E-09	8.45235E-12	2.94347E-13	3.60809E-16	-2.58250E-17
0.7	-3.67350E-08	8.44596E-10	-1.27320E-11	1.19515E-13	-6.25560E-16
0.8	-4.81590E-09	2.63934E-11	2.41904E-13	-5.36000E-15	3.73993E-17
0.9	4.84996E-08	-1.11690E-09	1.57464E-11	-1.35440E-13	6.54802E-16
1	-1.46850E-08	2.79200E-10	-3.87640E-12	3.48700E-14	-1.78540E-16

μ^*	C_{11}	C_{12}	C_{13}	C_{14}	C_{15}
0.05	0.00000E+00	-6.11957E-02	4.96171E-03	-1.55160E-04	1.81866E-06
0.1	0.00000E+00	-5.41190E-02	2.12818E-03	-4.49070E-05	4.21399E-07
0.15	0.00000E+00	-5.41630E-02	1.83445E-03	-2.83810E-05	1.56135E-07
0.2	6.17870E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.25	-1.74160E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.3	3.78562E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.333	0.00000E+00	-3.32979E-02	9.89014E-04	-2.00760E-05	2.13657E-07
0.4	0.00000E+00	-2.63120E-02	6.77611E-04	-1.18360E-05	1.19749E-07
0.5	1.39801E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.6	9.98442E-20	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.7	1.39E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.8	-9.40150E-20	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.9	-1.36400E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
1	3.96745E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00

Table 5.E.13 – Stress Factor K_{xy} Coefficients – Triangular Hole Pattern

μ^*	C_{16}				
0.05	-6.24160E-09				
0.1	-9.84060E-10				
0.15	2.71970E-10				
0.2	0.00000E+00				
0.25	0.00000E+00				
0.3	0.00000E+00				
0.333	-8.79980E-10				
0.4	-4.63550E-10				
0.5	0.00000E+00				
0.6	0.00000E+00				
0.7	0.00000E+00				
0.8	0.00000E+00				
0.9	0.00000E+00				
1	0.00000E+00				

Notes:

1. $$K_{xy} = \frac{C_1 + C_2\theta + C_3\theta^2 + C_4\theta^3 + C_5\theta^4 + C_6\theta^5 + C_7\theta^6 + C_8\theta^7 + C_9\theta^8 + C_{10}\theta^9 + C_{11}\theta^{10}}{1 + C_{12}\theta + C_{13}\theta^2 + C_{14}\theta^3 + C_{15}\theta^4 + C_{16}\theta^5}$$
2. See Figure 5.E.4 for definition of K_{xy} and θ (θ is in Degrees, $0 \le \theta \le 90$)
3. Valid range is $0.1 \le \mu^* \le 1.0$, data for $\mu^* = 0.05$ is provided for information only.

Table 5.E.14 – Stress Factor K_{xz} Coefficients – Triangular Hole Pattern

μ^*	C_1	C_2	C_3	C_4	C_5
0.05	-9.29310E-06	-2.40230E-05	5.42804E-06	-1.72080E-06	3.22669E-08
0.1	-9.93840E-06	-8.18720E-04	1.65045E-05	-4.11550E-06	8.28294E-08
0.15	-6.53010E-06	-3.54095E-03	1.29674E-04	-1.15250E-05	2.09898E-07
0.2	-3.61560E-06	-7.06114E-03	2.25220E-04	-1.50590E-05	2.53461E-07
0.25	2.74788E-06	-1.06818E-02	-8.15910E-06	-6.69010E-06	-1.84370E-07
0.3	1.04406E-06	-1.36716E-02	-4.17830E-05	-2.20820E-07	-3.72820E-07
0.333	-1.38930E-06	-1.55881E-02	5.74404E-05	2.27610E-06	-1.01860E-08
0.4	3.13558E-06	-1.85694E-02	1.62032E-05	-3.46980E-06	9.27417E-08
0.5	-3.16530E-06	-2.18793E-02	2.88419E-05	-3.40110E-06	2.55929E-07
0.6	5.79860E-06	-2.42588E-02	-9.32380E-06	3.04863E-06	-1.99940E-07
0.7	9.28851E-06	-2.60522E-02	-1.68180E-04	2.26662E-05	-1.42040E-06
0.8	-2.46430E-06	-2.92418E-02	8.80912E-05	-8.59660E-06	5.62546E-07
0.9	-9.50950E-07	-3.19965E-02	9.74574E-05	-9.48760E-06	6.50771E-07
1	7.19047E-05	-3.65095E-02	5.74262E-04	-6.81290E-05	4.06484E-06
μ^*	C_6	C_7	C_8	C_9	C_{10}
0.05	-1.69180E-10	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.1	-4.76560E-10	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.15	-1.18870E-09	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.2	-1.43010E-09	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.25	1.57697E-08	-4.29970E-10	7.54160E-12	-8.49970E-14	5.28050E-16
0.3	1.85814E-08	-4.27470E-10	6.37333E-12	-6.30240E-14	3.60467E-16
0.333	-3.88610E-11	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.4	-1.95120E-09	6.35820E-11	-1.12390E-12	9.39092E-15	-3.50260E-17
0.5	-1.10830E-08	3.19647E-10	-5.61770E-12	5.80545E-14	-3.26310E-16
0.6	1.00071E-08	-2.81100E-10	4.70851E-12	-4.66620E-14	2.51901E-16
0.7	5.53414E-08	-1.32840E-09	1.98654E-11	-1.80260E-13	9.07814E-16
0.8	-1.75270E-08	3.17054E-10	-3.27550E-12	1.70610E-14	-2.46830E-17
0.9	-2.27660E-08	5.02323E-10	-7.09120E-12	6.20570E-14	-3.06570E-16
1	-1.31660E-07	2.53689E-09	-2.94940E-11	1.99428E-13	-6.98860E-16
μ^*	C_{11}	C_{12}	C_{13}	C_{14}	C_{15}
0.05	0.00000E+00	-8.88523E-02	3.57223E-03	-7.14600E-05	6.88731E-07
0.1	0.00000E+00	-6.91433E-02	2.51059E-03	-4.68590E-05	4.27074E-07
0.15	0.00000E+00	-4.77362E-02	1.62971E-03	-2.86370E-05	2.42889E-07
0.2	0.00000E+00	-3.67894E-02	1.24346E-03	-2.08690E-05	1.72584E-07
0.25	-1.34680E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.3	-8.76600E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.333	0.00000E+00	-2.49274E-03	-6.12320E-04	1.55087E-05	-1.58170E-07
0.4	4.17428E-20	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.5	7.71866E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.6	-5.70300E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.7	-1.94600E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.8	-7.44600E-20	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.9	6.53218E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
1	9.11009E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00

Table 5.E.14 – Stress Factor K_{xz} Coefficients – Triangular Hole Pattern

μ^*	C_{16}				
0.05	-2.49920E-09				
0.1	-1.42360E-09				
0.15	-6.24940E-10				
0.2	-3.32030E-10				
0.25	0.00000E+00				
0.3	0.00000E+00				
0.333	5.79203E-10				
0.4	0.00000E+00				
0.5	0.00000E+00				
0.6	0.00000E+00				
0.7	0.00000E+00				
0.8	0.00000E+00				
0.9	0.00000E+00				
1	0.00000E+00				

Notes:

1. $$K_{xz} = \frac{C_1 + C_2\theta + C_3\theta^2 + C_4\theta^3 + C_5\theta^4 + C_6\theta^5 + C_7\theta^6 + C_8\theta^7 + C_9\theta^8 + C_{10}\theta^9 + C_{11}\theta^{10}}{1 + C_{12}\theta + C_{13}\theta^2 + C_{14}\theta^3 + C_{15}\theta^4 + C_{16}\theta^5}$$
2. See Figure 5.E.4 for definition of K_{xz} and θ (θ is in Degrees, $0 \le \theta \le 90$)
3. Valid range is $0.1 \le \mu^* \le 1.0$, data for $\mu^* = 0.05$ is provided for information only.

Table 5.E.15 – Stress Factor K_{yz} Coefficients – Triangular Hole Pattern

μ^*	C_1	C_2	C_3	C_4	C_5
0.05	1.01759E+00	-3.05131E-02	-2.78780E-04	2.03038E-05	-1.95910E-07
0.1	1.03660E+00	0.00000E+00	-4.75940E-04	0.00000E+00	1.16120E-07
0.15	1.05819E+00	0.00000E+00	-2.90970E-04	0.00000E+00	9.78866E-08
0.2	1.08253E+00	0.00000E+00	-2.23930E-04	0.00000E+00	1.07827E-07
0.25	1.10950E+00	4.96099E-05	-5.02180E-04	-1.98140E-06	5.75502E-07
0.3	1.14000E+00	-2.73920E-05	-3.62530E-04	-6.23200E-06	6.95315E-07
0.333	1.16250E+00	-2.10030E-04	-2.75170E-04	-8.95280E-06	7.13417E-07
0.4	1.21360E+00	-2.84640E-04	-1.95400E-04	-9.99600E-06	7.08864E-07
0.5	1.30250E+00	-1.43900E-04	-2.00590E-04	-2.55510E-06	1.10218E-07
0.6	1.40640E+00	1.24641E-04	-2.51630E-04	2.42722E-06	-8.75060E-08
0.7	1.52461E+00	3.88729E-05	-2.36130E-04	9.04305E-07	-1.24720E-07
0.8	1.66000E+00	-3.42500E-04	-1.40620E-04	-1.24210E-05	6.95292E-07
0.9	1.81620E+00	2.57972E-04	-3.23520E-04	5.03588E-06	-3.24240E-07
1	1.99998E+00	-4.08690E-04	-1.78600E-04	-1.48060E-05	9.30412E-07
μ^*	C_6	C_7	C_8	C_9	C_{10}
0.05	3.45278E-10	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.1	0.00000E+00	6.24853E-11	0.00000E+00	-1.80420E-14	0.00000E+00
0.15	0.00000E+00	3.81940E-11	0.00000E+00	-8.05400E-15	0.00000E+00
0.2	0.00000E+00	9.71070E-12	0.00000E+00	-2.67240E-15	0.00000E+00
0.25	-1.94130E-08	4.82249E-10	-8.70230E-12	9.41449E-14	-5.33350E-16
0.3	-2.45770E-08	6.01127E-10	-1.00750E-11	1.02627E-13	-5.62150E-16
0.333	-2.28300E-08	4.99708E-10	-7.60450E-12	7.21191E-14	-3.73300E-16
0.4	-2.43360E-08	5.54380E-10	-8.38470E-12	7.82984E-14	-4.03120E-16
0.5	-2.17130E-10	-6.35380E-11	1.67167E-12	-2.03770E-14	1.26185E-16
0.6	1.65579E-09	8.28289E-12	-8.55000E-13	1.42028E-14	-1.02380E-16
0.7	7.97510E-09	-2.53680E-10	4.60393E-12	-4.82230E-14	2.71723E-16
0.8	-2.18310E-08	4.14476E-10	-4.70290E-12	3.00729E-14	-9.16970E-17
0.9	1.33468E-08	-3.36510E-10	5.33822E-12	-5.18350E-14	2.81329E-16
1	-3.45480E-08	8.17163E-10	-1.23080E-11	1.14381E-13	-5.97320E-16
μ^*	C_{11}	C_{12}	C_{13}	C_{14}	C_{15}
0.05	0.00000E+00	-3.04984E-02	2.88546E-03	-1.07790E-04	1.51472E-06
0.1	1.13734E-18	0.00000E+00	9.03802E-04	0.00000E+00	-5.17010E-07
0.15	2.65237E-19	0.00000E+00	5.85740E-04	0.00000E+00	-1.71680E-07
0.2	0.00000E+00	0.00000E+00	3.96930E-04	0.00000E+00	-5.73760E-08
0.25	1.22126E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.3	1.27005E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.333	8.01462E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.4	8.72462E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.5	-3.18710E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.6	2.82046E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.7	-6.38360E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.8	7.44602E-20	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.9	-6.53220E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
1	1.33972E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00

Table 5.E.15 – Stress Factor K_{yz} Coefficients – Triangular Hole Pattern

μ^*	C_{16}	C_{17}	C_{18}	C_{19}	
0.05	-6.83930E-09	0.00000E+00	0.00000E+00	0.00000E+00	
0.1	0.00000E+00	1.70274E-10	-1.81320E-14	4.14493E-19	
0.15	0.00000E+00	7.75559E-11	-6.99310E-15	3.38942E-19	
0.2	0.00000E+00	3.85892E-11	-3.13170E-15	2.30988E-19	
0.25	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.3	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.333	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.4	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.5	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.6	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.7	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.8	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.9	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
1	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	

Notes:

1. $$K_{yz} = \frac{C_1 + C_2\theta + C_3\theta^2 + C_4\theta^3 + C_5\theta^4 + C_6\theta^5 + C_7\theta^6 + C_8\theta^7 + C_9\theta^8 + C_{10}\theta^9 + C_{11}\theta^{10}}{1 + C_{12}\theta + C_{13}\theta^2 + C_{14}\theta^3 + C_{15}\theta^4 + C_{16}\theta^5 + C_{17}\theta^6 + C_{18}\theta^8 + C_{19}\theta^{10}}$$
2. See Figure 5.E.4 for definition of K_y and θ (θ is in Degrees, $0 \le \theta \le 90$)
3. Valid range is $0.1 \le \mu^* \le 1.0$, data for $\mu^* = 0.05$ is provided for information only.

Table 5.E.16 – Stress Factors K_x And K_y Coefficients – Rectangular Hole Pattern

μ^*	C_1	C_2	C_3	C_4	C_5
0.05	-1.47580E-05	1.87249E-03	-7.61080E-04	1.08786E-04	-7.15640E-06
0.1	9.61530E-05	1.49451E-03	-6.65180E-04	1.10521E-04	-8.80780E-06
0.15	4.52677E-04	7.25376E-04	-3.88940E-04	6.22683E-05	-5.84370E-06
0.2	-4.59662E-03	4.14890E-04	-3.50340E-04	3.59267E-05	-3.37600E-06
0.25	-2.32447E-02	1.14393E-04	-2.62490E-04	1.07287E-05	-9.18750E-07
0.3	-5.81960E-02	9.31334E-05	-2.19360E-04	6.95148E-06	-2.82680E-07
0.4	-1.67582E-01	0.00000E+00	7.67000E-06	0.00000E+00	3.12582E-07
0.5	-3.04997E-01	2.44227E-04	2.90749E-04	1.03388E-05	-6.09000E-07
0.6	-4.51190E-01	1.82289E-04	5.63428E-04	5.55595E-06	-3.11900E-07
0.7	-5.99893E-01	-1.25650E-04	7.97795E-04	3.53441E-06	-4.01310E-07
0.8	-7.47205E-01	4.48376E-04	8.14352E-04	1.59400E-05	-1.01130E-06
0.9	-8.84697E-01	-5.33590E-05	1.07553E-03	5.57335E-06	-6.58380E-07
1	-1.00000E+00	-6.01550E-04	1.40134E-03	-2.02820E-05	1.00488E-06
μ^*	C_6	C_7	C_8	C_9	C_{10}
0.05	2.33550E-07	-3.74710E-09	2.27673E-11	1.04903E-13	-2.01210E-15
0.1	3.52872E-07	-7.55790E-09	8.91862E-11	-5.51610E-13	1.43371E-15
0.15	2.70724E-07	-6.52480E-09	8.77846E-11	-6.62150E-13	2.58130E-15
0.2	1.75074E-07	-4.56500E-09	6.55342E-11	-5.31400E-13	2.28801E-15
0.25	6.90285E-08	-2.06050E-09	3.07767E-11	-2.48780E-13	1.04292E-15
0.3	3.61104E-08	-1.24330E-09	1.96093E-11	-1.63430E-13	7.05734E-16
0.4	0.00000E+00	-2.94550E-11	0.00000E+00	0.00000E+00	0.00000E+00
0.5	2.93634E-08	-8.08080E-10	1.21405E-11	-1.02700E-13	4.65426E-16
0.6	9.48432E-09	-1.97890E-10	2.10909E-12	-9.14970E-15	-5.90480E-18
0.7	1.46249E-08	-3.61090E-10	5.25001E-12	-4.39820E-14	1.97926E-16
0.8	3.05689E-08	-6.27430E-10	8.21919E-12	-6.62010E-14	3.01974E-16
0.9	2.64004E-08	-7.20270E-10	1.19106E-11	-1.16290E-13	6.21431E-16
1	-3.55490E-08	6.67355E-10	-7.22950E-12	4.25909E-14	-1.05160E-16
μ^*	C_{11}	C_{12}	C_{13}	C_{14}	C_{15}
0.05	7.12847E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.1	-2.92300E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.15	-3.93740E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.2	-4.06880E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.25	-1.77570E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.3	-1.25400E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.4	0.00000E+00	3.66644E-04	1.39889E-08	4.71280E-12	-9.52780E-16
0.5	-8.85620E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.6	1.17895E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.7	-3.69860E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.8	-6.00950E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.9	-1.40370E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
1	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00

Table 5.E.16 – Stress Factors K_x And K_y Coefficients – Rectangular Hole Pattern

Notes:

1. $$K_x,\, K_y = \frac{C_1 + C_2\theta + C_3\theta^2 + C_4\theta^3 + C_5\theta^4 + C_6\theta^5 + C_7\theta^6 + C_8\theta^7 + C_9\theta^8 + C_{10}\theta^9 + C_{11}\theta^{10}}{1 + C_{12}\theta^2 + C_{13}\theta^4 + C_{14}\theta^6 + C_{15}\theta^8}$$
2. See Figure 5.E.4 for definition of K_x and K_y and θ (θ is in Degrees, $0 \le \theta \le 90$)
3. Valid range is $0.1 \le \mu^* \le 1.0$, data for $\mu^* = 0.05$ is provided for information only.

Table 5.E.17 – Stress Factor K_{xy} – Square Hole Pattern

μ^*	C_1	C_2	C_3	C_4	C_5
0.05	4.10628E-04	-1.03092E+00	-2.60562E-02	1.42763E-02	-1.13154E-03
0.1	-1.82830E-04	-4.93550E-01	-1.94314E-02	4.90571E-03	-2.95940E-04
0.15	-5.94810E-05	-3.51560E-01	-4.36757E-03	1.34861E-03	-5.37530E-05
0.2	5.45892E-04	-2.81094E-01	1.23677E-02	-2.05420E-04	1.14121E-06
0.25	1.32829E-05	-2.22708E-01	-2.00720E-04	2.29490E-04	1.05088E-06
0.3	6.15483E-06	-1.89244E-01	2.07521E-04	9.47928E-05	5.03027E-06
0.4	1.66864E-06	-1.48510E-01	4.61787E-05	6.25567E-05	9.48205E-07
0.5	-1.20220E-06	-1.28243E-01	2.58038E-03	-2.56740E-05	1.42611E-07
0.6	-4.63660E-06	-1.19644E-01	-3.38840E-05	3.23761E-05	-2.36380E-07
0.7	4.34520E-06	-1.18318E-01	3.89048E-05	1.89838E-05	4.09430E-07
0.8	6.30422E-06	-1.21221E-01	1.89750E-06	2.05803E-05	4.80584E-07
0.9	-4.40310E-07	-1.28514E-01	9.10516E-05	1.44325E-05	7.77422E-07
1	-5.06060E-06	-1.39930E-01	7.41931E-05	2.11002E-05	3.89961E-07
μ^*	C_6	C_7	C_8	C_9	C_{10}
0.05	4.58608E-05	-1.11640E-06	1.69499E-08	-1.57290E-10	8.16374E-13
0.1	9.90563E-06	-2.11290E-07	2.94076E-09	-2.59170E-11	1.31470E-13
0.15	1.01724E-06	-1.01090E-08	3.65386E-11	2.29882E-13	-2.45460E-15
0.2	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.25	-3.33450E-07	9.60689E-09	-1.41310E-10	1.24782E-12	-6.33550E-15
0.3	-3.62510E-07	9.01762E-09	-1.24960E-10	1.06743E-12	-5.33570E-15
0.4	-8.06470E-08	1.46557E-09	-1.09650E-11	2.84023E-14	1.52304E-17
0.5	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.6	4.87650E-09	-2.33200E-10	5.13313E-12	-5.70940E-14	3.17302E-16
0.7	-1.81480E-08	3.88068E-10	-5.54280E-12	4.99395E-14	-2.55870E-16
0.8	-2.62660E-08	6.99127E-10	-1.15610E-11	1.13080E-13	-5.99740E-16
0.9	-3.24290E-08	7.59367E-10	-1.16550E-11	1.09684E-13	-5.72670E-16
1	-1.46850E-08	2.79200E-10	-3.87640E-12	3.48700E-14	-1.78540E-16
μ^*	C_{11}	C_{12}	C_{13}	C_{14}	C_{15}
0.05	-1.81420E-15	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.1	-2.92160E-16	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.15	5.45478E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.2	0.00000E+00	-3.78601E-02	1.24548E-03	-1.83300E-05	1.01840E-07
0.25	1.40788E-17	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.3	1.18572E-17	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.4	-3.38450E-20	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.5	0.00000E+00	-2.00096E-02	4.98896E-04	-6.14640E-06	3.41459E-08
0.6	-7.05110E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.7	5.68605E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.8	1.33276E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.9	1.27259E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
1	3.96745E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00

Table 5.E.17 – Stress Factor K_{xy} – Square Hole Pattern

Notes:

1. $$K_{xy} = \frac{C_1 + C_2\theta + C_3\theta^2 + C_4\theta^3 + C_5\theta^4 + C_6\theta^5 + C_7\theta^6 + C_8\theta^7 + C_9\theta^8 + C_{10}\theta^9 + C_{11}\theta^{10}}{1 + C_{12}\theta + C_{13}\theta^2 + C_{14}\theta^3 + C_{15}\theta^4}$$
2. See Figure 5.E.4 for definition of K_{xy} and θ (θ is in Degrees, $0 \le \theta \le 90$)
3. Valid range is $0.1 \le \mu^* \le 1.0$, data for $\mu^* = 0.05$ is provided for information only.

Table 5.E.18 – Stress Factor K_{xz} And K_{yz} – Square Hole Pattern

μ^*	C_1	C_2	C_3	C_4	C_5
0.05	-8.65900E-05	0.00000E+00	3.39079E-06	0.00000E+00	-9.00340E-08
0.1	-9.16630E-06	1.70502E-04	-2.18240E-04	3.03915E-05	-2.60740E-06
0.15	-7.50710E-06	-1.70443E-03	6.97716E-06	-6.04360E-06	8.60699E-08
0.2	5.53257E-07	-3.89704E-03	-1.93130E-05	-2.30990E-06	-3.09000E-07
0.25	3.23710E-07	-6.44426E-03	-6.52400E-05	2.93245E-06	-5.32400E-07
0.3	-6.76800E-07	-9.59777E-03	3.72356E-05	-8.93710E-06	3.33448E-07
0.4	2.53798E-06	-1.47723E-02	-2.99370E-05	2.92311E-06	-3.37840E-07
0.5	8.12376E-06	-1.93888E-02	2.78977E-05	-3.32890E-06	2.15640E-07
0.6	3.52335E-06	-2.25507E-02	-1.01380E-04	1.34464E-05	-8.10840E-07
0.7	7.44733E-06	-2.58912E-02	-2.05060E-05	4.19794E-06	-2.12710E-07
0.8	1.25243E-05	-2.86525E-02	4.06228E-06	-3.16500E-06	5.12292E-07
0.9	-9.44320E-06	-3.16364E-02	-3.74000E-05	8.62779E-06	-5.59110E-07
1	-1.97550E-05	-3.43965E-02	-1.45890E-04	1.85432E-05	-1.03270E-06
μ^*	C_6	C_7	C_8	C_9	C_{10}
0.05	0.00000E+00	1.39181E-11	0.00000E+00	-2.65320E-15	0.00000E+00
0.1	1.08751E-07	-2.67100E-09	4.13442E-11	-4.00270E-13	2.20859E-15
0.15	-5.10320E-10	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.2	1.60946E-08	-3.29790E-10	3.54744E-12	-2.20680E-14	8.74181E-17
0.25	2.31330E-08	-4.96950E-10	6.17680E-12	-4.62380E-14	2.01806E-16
0.3	-1.09310E-08	3.01384E-10	-5.42280E-12	5.71883E-14	-3.19690E-16
0.4	1.47598E-08	-3.45240E-10	4.84820E-12	-4.08380E-14	1.90865E-16
0.5	-8.79080E-09	2.38954E-10	-4.04210E-12	4.10517E-14	-2.29180E-16
0.6	2.96526E-08	-6.57410E-10	9.03011E-12	-7.49820E-14	3.44094E-16
0.7	8.54418E-09	-2.09120E-10	3.16043E-12	-2.87240E-14	1.43406E-16
0.8	-2.58810E-08	7.24790E-10	-1.20140E-11	1.17377E-13	-6.25690E-16
0.9	2.39629E-08	-6.08270E-10	9.39984E-12	-8.70290E-14	4.43996E-16
1	3.80915E-08	-8.85470E-10	1.30648E-11	-1.18880E-13	6.08423E-16
μ^*	C_{11}	C_{12}	C_{13}	C_{14}	C_{15}
0.05	1.37550E-19	0.00000E+00	1.79422E-05	0.00000E+00	4.15212E-07
0.1	-5.25320E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.15	0.00000E+00	2.23725E-03	-9.03060E-04	3.77631E-05	-5.40970E-07
0.2	-1.98940E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.25	-4.09670E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.3	7.31063E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.4	-3.82080E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.5	5.40118E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.6	-6.68450E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.7	-3.01410E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.8	1.40496E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
0.9	-9.60370E-19	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00
1	-1.33970E-18	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00

Table 5.E.18 – Stress Factor K_{xz} And K_{yz} – Square Hole Pattern

μ^*	C_{16}	C_{17}	C_{18}	C_{19}	
0.05	0.00000E+00	-9.82930E-11	7.22485E-15	-6.39790E-20	
0.1	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.15	2.68316E-09	0.00000E+00	0.00000E+00	0.00000E+00	
0.2	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.25	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.3	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.4	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.5	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.6	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.7	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.8	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
0.9	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	
1	0.00000E+00	0.00000E+00	0.00000E+00	0.00000E+00	

Notes:

1. $$K_{xz},\ K_{yz} = \frac{C_1 + C_2\theta + C_3\theta^2 + C_4\theta^3 + C_5\theta^4 + C_6\theta^5 + C_7\theta^6 + C_8\theta^7 + C_9\theta^8 + C_{10}\theta^9 + C_{11}\theta^{10}}{1 + C_{12}\theta + C_{13}\theta^2 + C_{14}\theta^3 + C_{15}\theta^4 + C_{16}\theta^5 + C_{17}\theta^6 + C_{18}\theta^8 + C_{19}\theta^{10}}$$
2. See Figure 5.E.4 for definition of K_{xz} and K_{yz} and θ (θ is in Degrees, $0 \le \theta \le 90$)
3. Valid range is $0.1 \le \mu^* \le 1.0$, data for $\mu^* = 0.05$ is provided for information only.

Table 5.E.19 – Boundary Conditions for the Numerical Analysis (see Figure 5.E.3)

Pressure Region	Applied Pressure
p_1	$p_t^* - p_h$
p_2	$p_s^* - p_h$
p_3	$p_s - p_h$
p_4	$-p_h$
p_5	$p_t - p_h$

Definitions

Tube Expansion Into Tubesheet	Tube-To-Tubesheet Weld Location	p_t^*	p_s^*	p_h
unexpanded	Tubeside	$p_t\left(1-q\dfrac{A_h}{A}\right)$	$p_s\left(1-q\dfrac{A_{ts}}{A}\right)$	p_s
	Shellside	$p_t\left(1-\dfrac{A_h}{A}\right)$	$p_s\left(1-q\dfrac{A_{ts}}{A}\right)-(q-1)\dfrac{A_h}{A}p_t$	p_t
expanded	Tubeside	$p_t\left(1-q\dfrac{A_h}{A}\right)$	$p_s\left(1-q\dfrac{A_{ts}}{A}\right)$	$\dfrac{t_{ts}}{h}p_t+\left(1-\dfrac{t_{ts}}{h}\right)p_s$
	Shellside	$p_t\left(1-\dfrac{A_h}{A}\right)$	$p_s\left(1-q\dfrac{A_{ts}}{A}\right)-(q-1)\dfrac{A_h}{A}p_t$	p_t

Notes:

1. $q = 0$ $\quad$ $(for\ the\ U-tube\ design)$
2. $q = 1$ $\quad$ $(for\ the\ Fixed\ Tubesheet\ design)$
3. $\dfrac{A_{ts}}{A} = f_p\left(\dfrac{d}{p}\right)^2$
4. $\dfrac{A_h}{A} = f_p\left(\dfrac{d-t_t}{p}\right)^2$
5. $f_p = \dfrac{\pi}{2\sqrt{3}}$ $\quad$ $(for\ the\ triangular\ pattern)$
6. $f_p = \dfrac{\pi}{4}$ $\quad$ $(for\ the\ square\ pattern)$

5.E.10 Figures

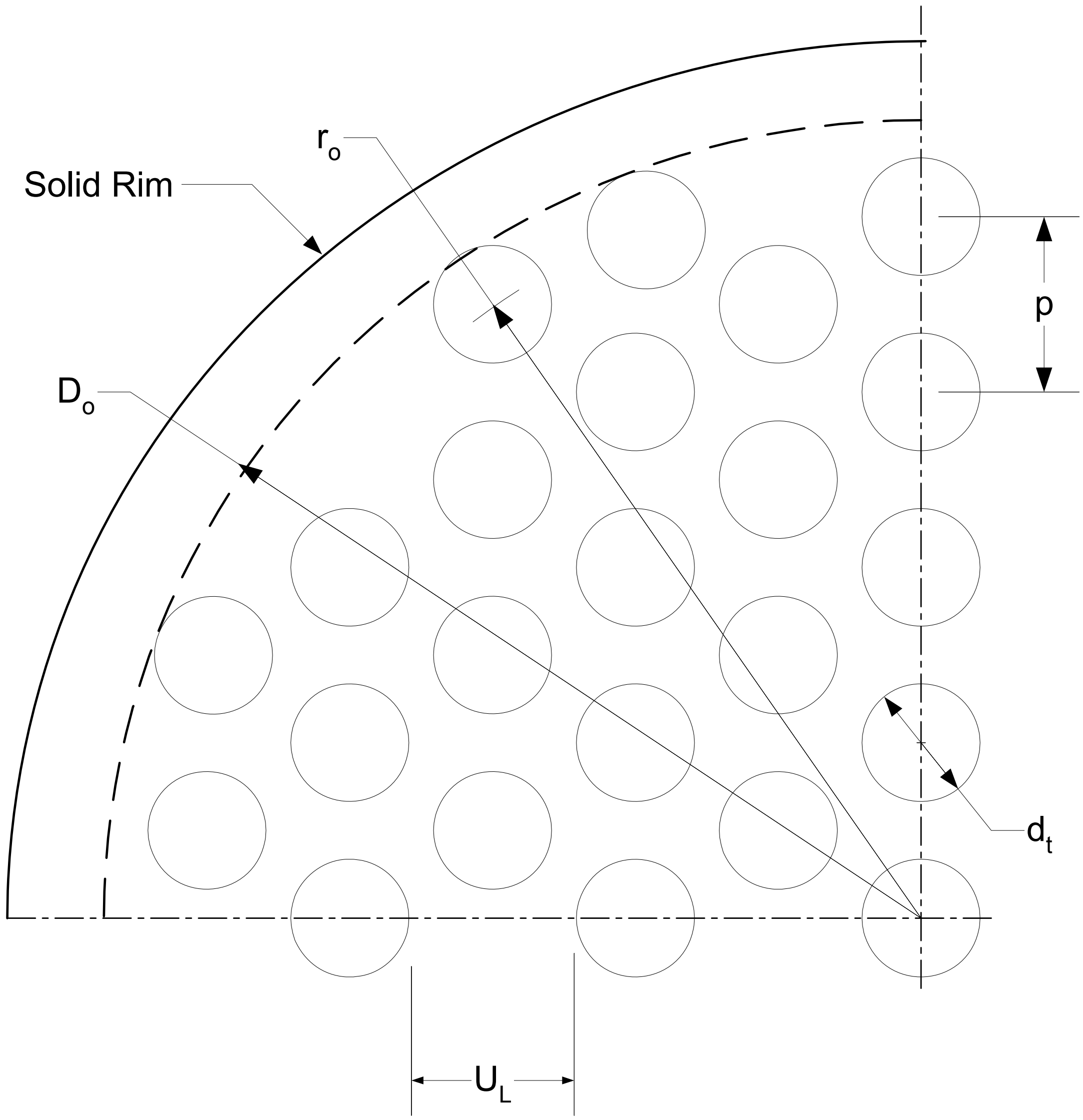
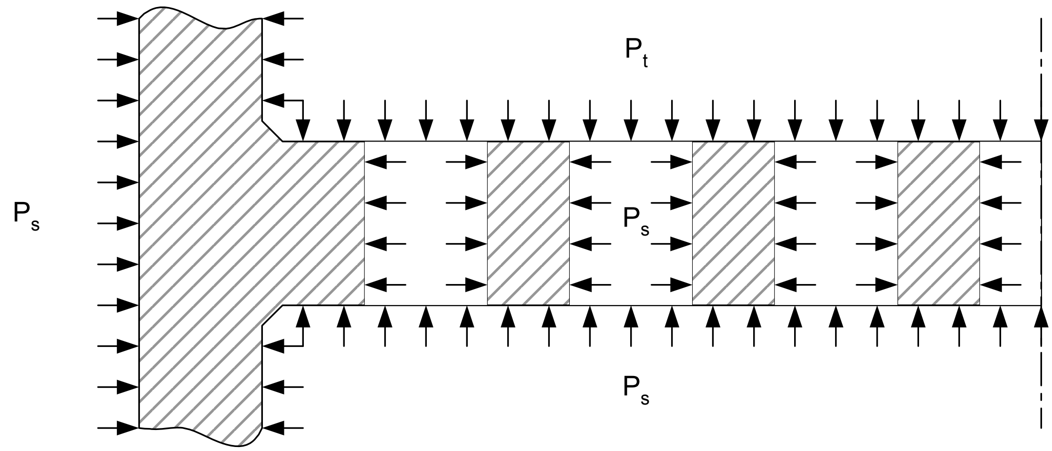
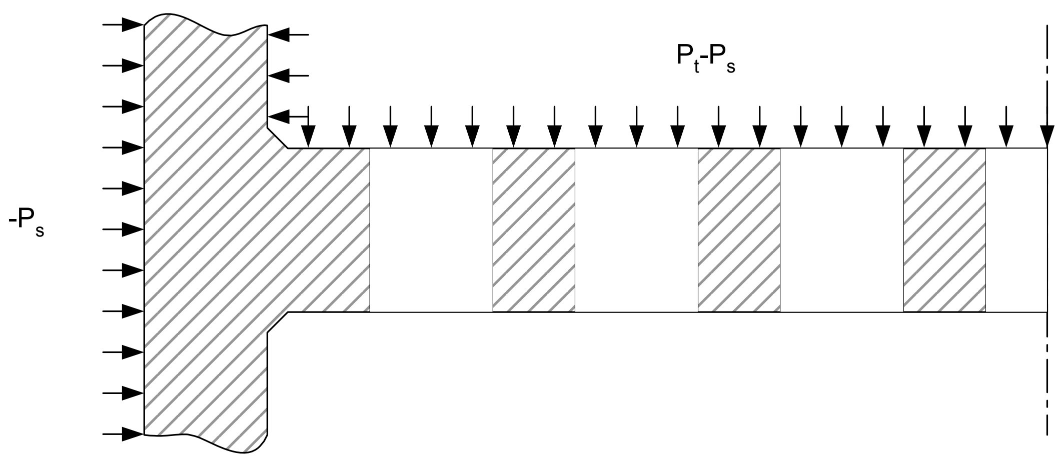


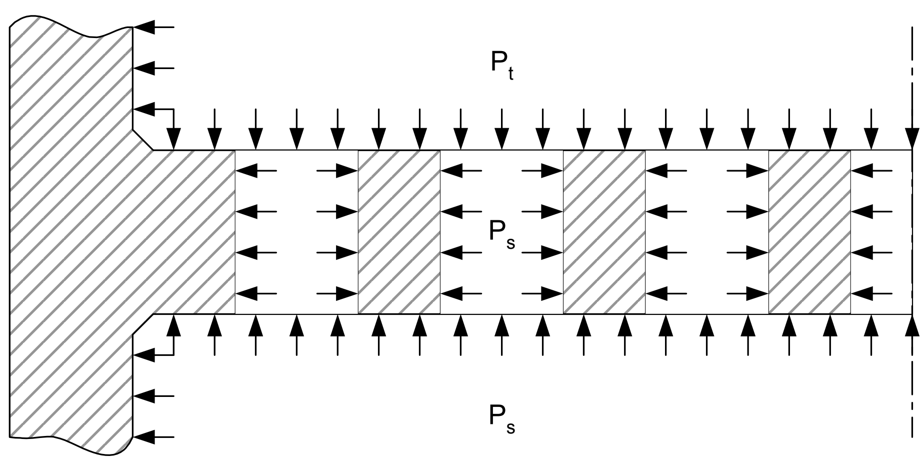
Figure 5.E.1
Perforated Plate Geometry Details



(A) Hydrostatic Compression

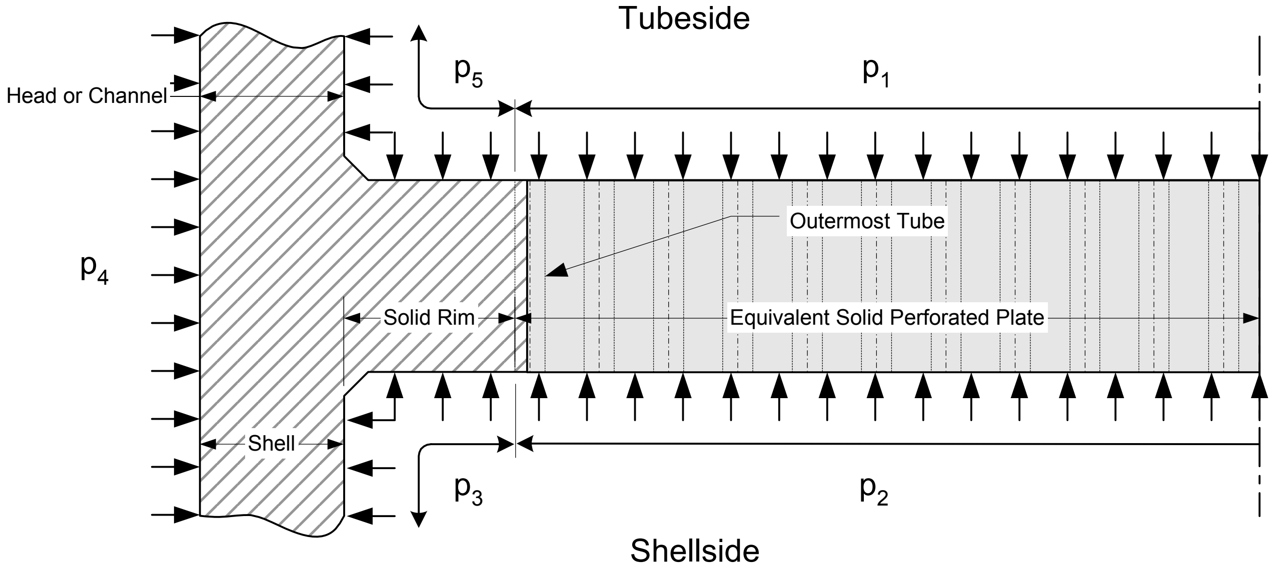


(B) Finite Element Analysis Boundary Condition



(C) Desired Loading Condition w/ Holes Under Shell Pressure

Figure 5.E.2
Perforated Plate Geometry Details



Note: See Table 5.19 for definition of boundary conditions.

Figure 5.E.3
Boundary Conditions for Numerical Analysis

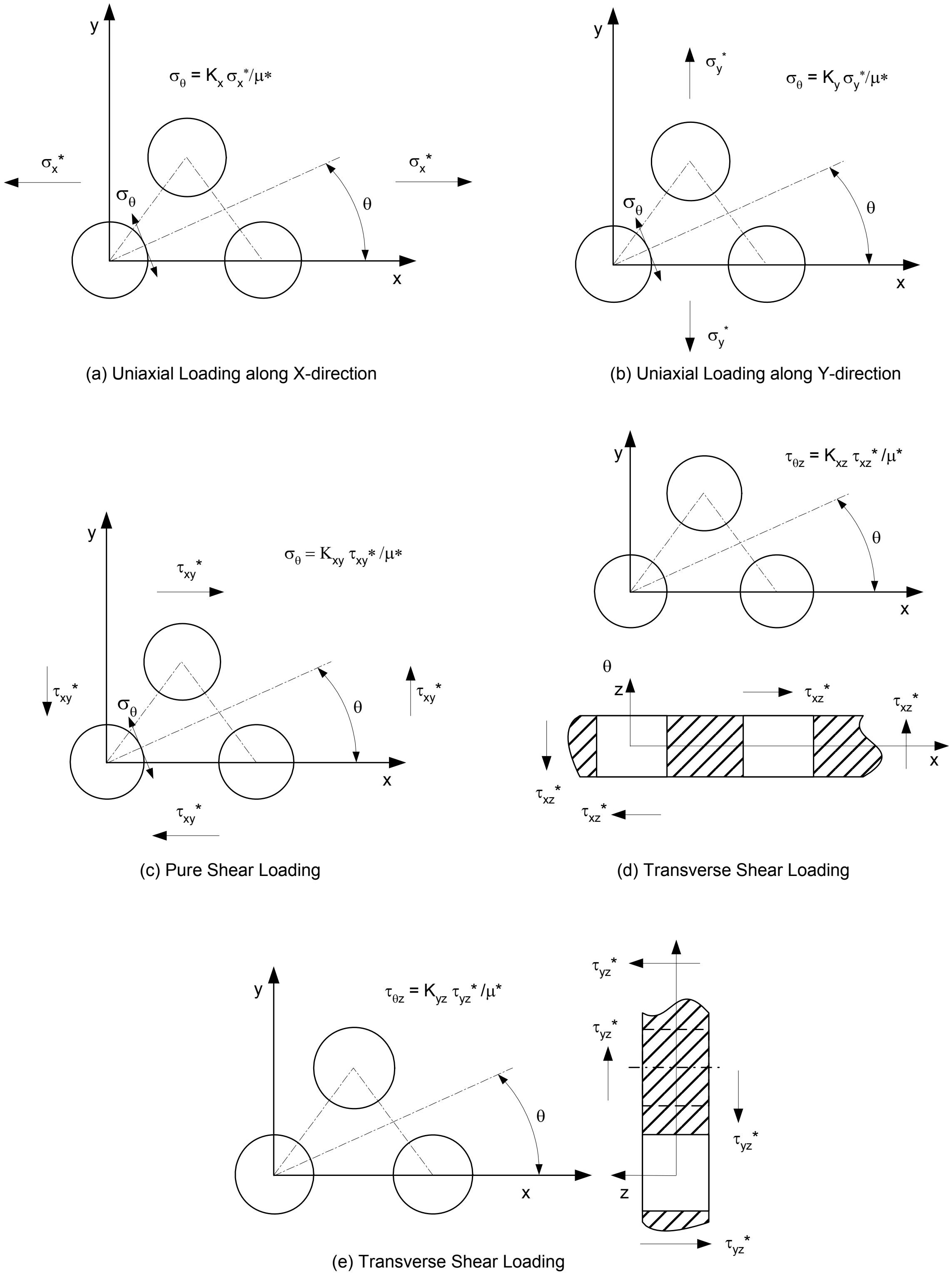
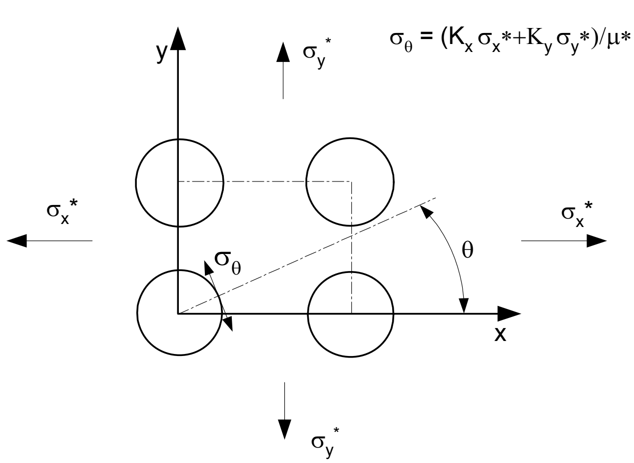
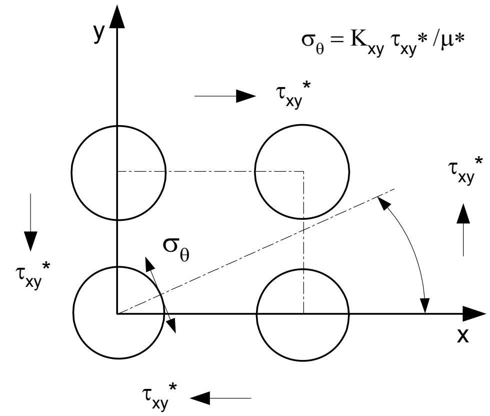


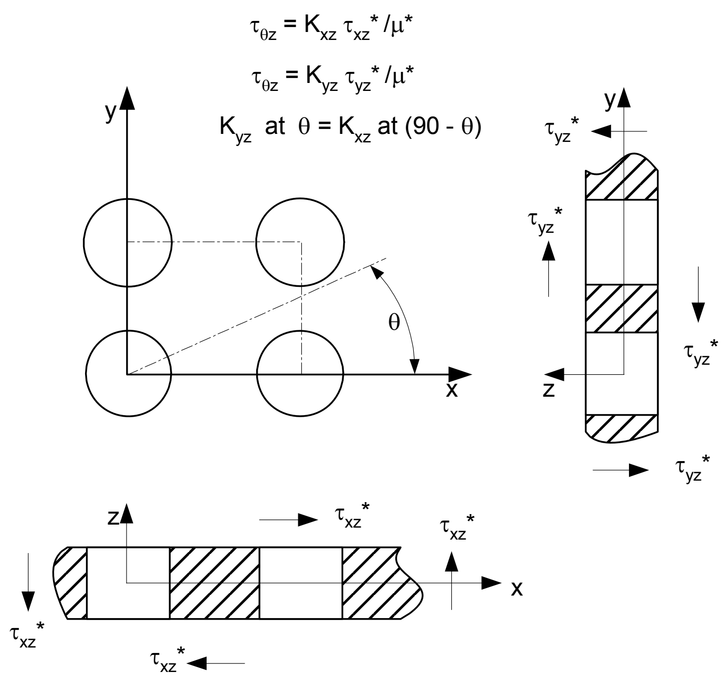
Figure 5.E.4
Stress Orientations For Perforated Plate With Triangular Pattern Holes



(a) Uniaxial Loading in Pitch Direction



(b) Pure Shear Loading



(c) Transverse Shear Loading

Figure 5.E.5
Stress Orientations For Perforated Plate With Square Pattern Holes

ANNEX 5.F

EXPERIMENTAL STRESS AND FATIGUE ANALYSIS

(NORMATIVE)

5.F.1 Overview

5.F.1.1 The critical or governing stresses in parts may be substantiated by experimental stress analysis. Experimental analysis may be used to determine governing stresses by strain measurement, and to evaluate the adequacy of a part for cyclic loading.

5.F.1.2 The test procedures followed and the interpretation of the results shall be such as to discount the effects of material added to the thickness of members, such as corrosion allowance or other material that cannot be considered as contributing to the strength of the part.

5.F.1.3 Re-evaluation is not required for configurations for which there are available detailed experimental results that are consistent with the requirements of this Annex.

5.F.1.4 The use of this Annex requires the approval of the user or an agent acting on behalf of the user and the acceptance of its use shall be documented in the Manufacturer's Design Report.

5.F.2 Strain Measurement Test Procedure for Stress Components

5.F.2.1 Permissible types of tests for the determination of governing stresses are strain measurement and photoelastic tests. Either two-dimensional or three-dimensional photoelastic techniques may be used as long as the model represents the structural effects of the loading.

5.F.2.2 Strain gages used may be of any type capable of indicating strains to 0.00005 in./in. (0.005%). It is recommended that the gage length be such that the maximum strain within the gage length does not exceed the average strain within the gage length by more than 10%. Instrumentation shall be such that both surface principal stresses may be determined at each gage location in the elastic range of material behavior at that gage location. A similar number and orientation of gages at each gage location are required to be used in tests beyond the elastic range of material behavior. The strain gages and cements that are used shall be shown to be reliable for use on the material surface finish, test temperature, and configuration considered to strain values at least 50% higher than those expected.

5.F.2.3 Strain gage data may be obtained from the actual component or from a model component of any scale that meets the gage length requirements defined in paragraph 5.F.2.2. The model material need not be the same as the component material but shall have an elastic modulus that is either known or has been measured at the test conditions. The requirements of dimensional similitude shall be met.

5.F.2.4 Sufficient locations on the vessel shall be investigated to ensure that measurements are taken at the most critical areas. The location of the critical areas and the optimum orientation of test gages may be determined by a brittle coating test.

5.F.2.5 Pressure gages shall meet the requirements of Part 8.

5.F.2.6 The internal pressure or mechanical load shall be applied in such increments that the variation of strain with load can be plotted so as to establish the ratio of stress to load in the elastic range. If the first loading results in strains that are not linearly proportional to the load, then it is permissible to unload and reload successively until the linear proportionality has been established. When frozen stress photoelastic techniques are used, only one load value can be applied in which case the load shall not be so high as to result in deformations that invalidate the test results. After all instrumentation has been deemed acceptable, the test should be continued on a strain or displacement controlled basis with adequate time permitted between load changes for all metal flow to be completed.

5.F.2.7 Linear elastic theory shall be used to determine the design load stresses from the strain gage data. The calculations shall be performed under the assumption that the material is elastic. The elastic constants used in the evaluation of experimental data shall be those applicable to the test material at the test temperature.

5.F.2.8 The extent of experimental stress analysis performed shall be sufficient to determine the governing stresses. When possible, combined analytical and experimental methods shall be used to distinguish between primary, secondary, and peak stresses.

5.F.2.9 Stress determined by experimental results shall be evaluated for protection against plastic collapse using the criterion in paragraph 5.2.2. Protection against cyclic loading shall be evaluated in accordance with paragraphs 5.5.3 or 5.F.3.

5.F.2.10 Tests conducted in accordance with this paragraph do not need to be witnessed by the Inspector. However, a detailed report of the test procedure and the results obtained shall be included with the Design Report. The Report shall show that the instrumentation used was within calibration.

5.F.3 Protection Against Cyclic Loading

5.F.3.1 The adequacy of a vessel or part to withstand cyclic loading may be demonstrated by means of a fatigue test in lieu of the methods of paragraph 5.5.3. The fatigue test shall not be used, however, as justification for exceeding the allowable values of primary or primary plus secondary stresses. This procedure shall not be used when the design temperature exceed the maximum temperature allowed for the fatigue curves as given in Annex 3.F. This procedure shall not be used when the number of design cycles exceed 50,000.

5.F.3.2 When a fatigue test is used to demonstrate the adequacy of a component to withstand cyclic loading, a description of the test shall be included in the Manufacturer's Design Report. This description shall contain sufficient detail to show compliance with the requirements of this Annex.

5.F.3.3 The test component or portion thereof shall be constructed of material having the same composition and subjected to mechanical working and heat treatment that result in mechanical properties equivalent to those of the material in the prototype component. Geometrical similarity must be maintained, at least in those portions whose ability to withstand cyclic loading is being investigated and in those adjacent areas which affect the stresses in the portion under test.

5.F.3.4 The test component or portion thereof shall withstand the number of cycles as set forth in paragraph 5.F.3.5 before failure occurs. Failure is herein defined as a propagation of a crack through the entire thickness, such as would produce a measurable leak in a pressure retaining member.

5.F.3.5 The minimum number of cycles, N_T, which the component must withstand, and the magnitude of the loading, L_T, to be applied to the component during test, shall be determined by multiplying the design service cycles N_D by a specified factor K_{TN} and the design service loads L_D by a specified factor K_{TS}.

Values of these factors shall be determined by means of the test parameter ratio diagram, the construction of which is as follows and is shown in Figure 5.F.1.

a) Project a vertical line from the design service cycles, N_D, on the abscissa of the S_a versus N diagram, until the line intersects the fatigue design curve, label this point S_{aD}. Determine the value of $K_s \cdot S_{aD}$ to establish point A. The parameter K_s is determined using paragraph 5.F.3.9.

b) Extend a horizontal line through the point D from the ordinate axis to an abscissa value of K_n times N_D. Label this point B. This parameter K_n is determined using paragraph 5.F.3.9.

c) Connect the points *A* and *B.* The segment *AB* embraces all the allowable combinations of K_{TS} and K_{TN}. Any point C on this segment may be chosen. The factors K_{TS} and K_{TN} are defined by the following equations (see Figure 5.F.1).

$$K_{TS} = \frac{S_{aC}}{S_{aD}} \tag{5.F.1}$$

$$K_{TN} = \frac{N_C}{N_D} \tag{5.F.2}$$

Therefore,

$$L_T = K_{TS} \cdot L_D \tag{5.F.3}$$

$$N_T = K_{TN} \cdot N_D \tag{5.F.4}$$

5.F.3.6 It should be noted that if the test article is not a full size component but a geometrically similar model, the value L_T would have to be adjusted by the appropriate scale factor, to be determined from structural similitude principles, if the loading is other than pressure. The number of cycles that the component must withstand during this test without failure must not be less than N_T, while subjected to a cyclic test loading L_T which shall be adjusted, if required, using model similitude principles if the component is not full size.

5.F.3.7 Accelerated fatigue testing (test cycles N_T are less than design cycles N_D) may be conducted if the design cycles N_D are greater than 10^4 and the testing conditions are determined by the following procedures which are illustrated in Figure 5.F.2. In this figure, the points A, B, and D correspond to similar labeled points in Figure 5.F.1.

a) The minimum number of test cycles $N_{T\min}$ is given by Equation (5.F.5). Project a vertical line through $N_{T\min}$ on the abscissa of the S_a versus N diagram such that it intersects and extends beyond the fatigue design curve.

$$N_{T\min} = 10^2 \sqrt{N_D} \tag{5.F.5}$$

b) Project a vertical line from the design service cycles, N_D, on the abscissa of the S_a versus N diagram, until the line intersects the fatigue design curve, label this point S_{aD}. Determine the value of $K_s \cdot S_{aD}$ to establish point A. The parameter K_s is determined using paragraph 5.F.3.9.

c) Construct a curve through the point *A* by multiplying all of the ordinate values (Sa values) on the fatigue design curve by the factor K_S. Label the intersection of this curve and a vertical line projection of $N_{T\min}$ as A'.

d) Any point *C* on the segment A', A, B determines the allowable combinations of K_{TS} and K_{TN}. The factors K_{TS} and K_{TN} are obtained in the same manner as in paragraph 5.F.3.5.

5.F.3.8 In certain instances, it may be desirable (or possible) in performing the test to increase only the loading or number of cycles, but not both, in which event two special cases of interest result from the above general case.

a) Case 1 (Factor Applied to Cycles Only) – In this case $K_{TS} = 1$ and the value of K_{TN} is determined by Equation (5.F.6). The number of test cycles that the component shall withstand during this test shall not be less than given by Equation (5.F.7) while subjected to the cyclic design service loading, adjusted as required, if a geometrically similar model is used.

$$K_{TN} = \frac{N_B}{N_D} \tag{5.F.6}$$

$$N_T = N_B = K_{TN} \cdot N_D \tag{5.F.7}$$

b) Case 2 (Factor Applied to Loading Only) – In this case $K_{TN} = 1$ and the value of K_{TS} is determined by Equation (5.F.8). The component shall withstand a number of cycles at least equal to the number of design service cycles given by Equation (5.F.9) while subjected to a cyclic test loading again adjusted as required, if a geometrically similar model is used.

$$K_{TS} = \frac{S_{aA}}{S_{aD}} \tag{5.F.8}$$

$$L_T = K_{TS} \cdot L_D \tag{5.F.9}$$

5.F.3.9 The values of K_n and K_s are the multiples of factors which account for the effects of size, surface finish, cyclic rate, temperature, and the number of replicate tests performed. They shall be determined as follows:

$$K_n = \max\left[(K_s)^{4.3},\ 2.6\right] \tag{5.F.10}$$

$$K_s = \max\left[\left(K_{sc} \cdot K_{sl} \cdot K_{sf} \cdot K_{st} \cdot K_{ss}\right),\ 1.25\right] \tag{5.F.11}$$

where:

$$K_{sc} = \max\left[\left(\frac{S_a\left(N,T_C\right)}{S_a\left(N,T_D\right)} \cdot \frac{S_{ae}\left(T_T\right)}{S_{ae}\left(T_C\right)}\right),\ 1.0\right] \tag{5.F.12}$$

$$K_{sl} = \max\left[\left(1.5 - 0.5R_{LP}\right),\ 1.0\right] \tag{5.F.13}$$

$$K_{sf} = \max\left[\left(1.175 - 0.175R_{SF}\right),\ 1.0\right] \tag{5.F.14}$$

$$K_{st} = \max\left[\left(\frac{S_a\left(N,T_T\right)}{S_a\left(N,T_D\right)}\right),\ 1.0\right] \tag{5.F.15}$$

$$K_{ss} = \max\left[\left(1.470 - 0.044N_{RT}\right),\ 1.0\right] \tag{5.F.16}$$

5.F.3.10 Experimental determination of fatigue strength reduction factors shall be in accordance with the following procedures.

a) The test part shall be fabricated from a material within the same P-Number grouping, and shall be subjected to the same heat treatment as the component.

b) The stress level in the specimen shall be such that the equivalent stress satisfies Equation (5.F.17) .

$$\left(P_L + P_b + Q\right) \le S_{PS} \tag{5.F.17}$$

c) The configuration, surface finish, and stress state of the specimen shall closely simulate those expected in the components. In particular, the stress gradient shall not be less abrupt than that expected in the component.

d) The cyclic rate shall be such that appreciable heating of the specimen does not occur.

e) The fatigue strength reduction factor shall preferably be determined by performing tests on notched and unnotched specimens, and calculated as the ratio of the unnotched stress to the notched stress for failure.

5.F.4 Nomenclature

K_n	multiplier to design cycles that accounts for the effects of size, surface finish, cyclic rate, temperature, and the number of replicate tests performed.
K_s	multiplier to design allowable stress that accounts for the effects of size, surface finish, cyclic rate, temperature, and the number of replicate tests performed.
K_{sc}	factor for differences in design fatigue curves at various temperatures.
K_{sf}	factor for the effect of surface finish.
K_{skin}	stress multiplier for thermal skin stress.
K_{sl}	factor for the effect of size on fatigue life.
K_{TN}	multiplier for test cycles.
K_{TS}	multiplier for test loading.
K_{ss}	factor for the statistical variation in test results.
K_{st}	factor for the effect of test temperature.
L_D	design service loading.
L_T	cyclic test loading.

N_B fatigue cycles at point B.

N_C fatigue cycles at point C.

N_D design cycles.

N_{RT} number of replicate tests.

N_T test cycles.

$N_{T\min}$ minimum number of test cycles.

P_b primary bending stress intensity.

P_L local primary membrane stress intensity.

Q secondary stress intensity.

R_{LP} ratio of model linear size to the linear size of the fabricated part

R_{SF} ratio of model surface finish to surface finish of the fabricated part

S_a alternating stress obtained from a fatigue curve for the specified number of operating cycles.

S_{aA} alternating stress at point A.

S_{aC} alternating stress at point C.

S_{aD} alternating stress at point D .

$S_a\left(N,T_C\right)$ stress amplitude from the applicable design fatigue curve (see Annex 3.F) for N cycles evaluated at T_C.

$S_a\left(N,T_D\right)$ stress amplitude from the applicable design fatigue curve (see Annex 3.F) for N cycles evaluated at the design temperature, T_D.

$S_a\left(N,T_T\right)$ stress amplitude from the applicable design fatigue curve (see Annex 3.F) for N cycles evaluated at the test temperature, T_T.

$S_{ae}\left(T_C\right)$ stress amplitude from the applicable design fatigue curve (see Annex 3.F) at the maximum number of cycles defined on the curve (possibly the endurance limit) evaluated at T_C.

$S_{ae}\left(T_T\right)$ stress amplitude from the applicable design fatigue curve (see Annex 3.F) at the maximum number of cycles defined on the curve (possibly the endurance limit) evaluated at the test temperature, T_T.

S_{as} stress from the applicable design fatigue curve (see Annex 3.F). If the design cycles is greater than 10^6, then S_{as} is determined from the fatigue curve at 10^6 cycles; otherwise, S_{as} is taken as the stress associated with the maximum number of cycles on the design fatigue curve.

T_C component temperature.

T_D design temperature.

T_T test temperature.

5.F.5 Figures

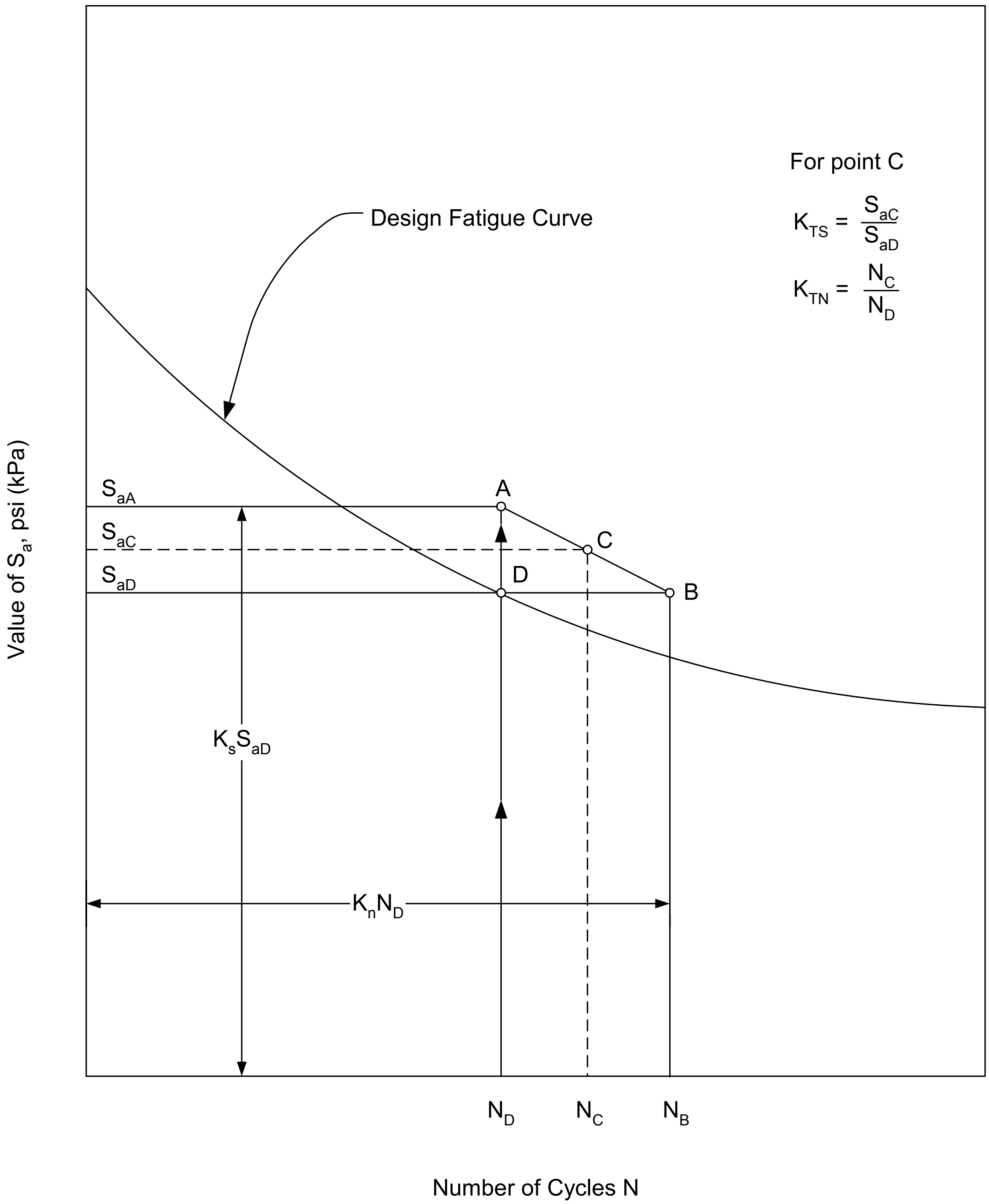


Figure 5.F.1
Construction Of The Testing Parameter Ratio Diagram

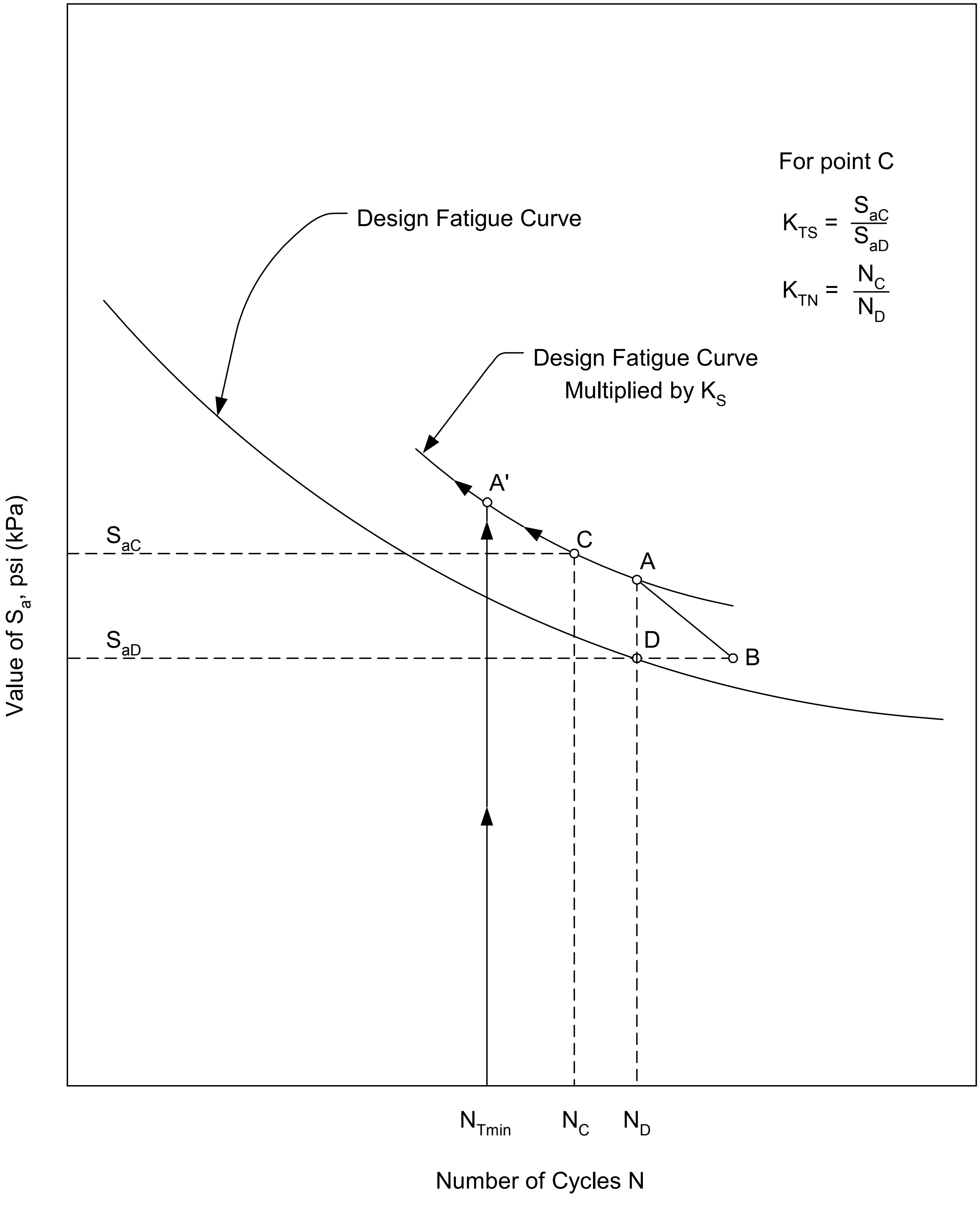


Figure 5.F.2
Construction Of The Testing Parameter Ratio Diagram For Accelerated Tests

PART 6

FABRICATION REQUIREMENTS

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6.1 General Fabrication Requirements

6.1.1 Materials

6.1.1.1 Documentation of Material Treatments, Tests, and Examination

The vessel Manufacturer shall document compliance with the special requirements of Part 3 for any of the treatments, tests, or examinations specified therein that are performed. The documentation shall include reports of the results of all tests and examinations performed on the materials by the vessel Manufacturer.

6.1.1.2 Material Identification

a) Material Marking – Material for pressure parts should be laid out so that when the vessel is completed, one complete set of the original identification markings required in the specifications for the material will be plainly visible. In case the original identification markings are unavoidably cut out or the material is divided into two or more parts, one set shall either be accurately transferred prior to cutting by the pressure vessel Manufacturer to a location where the markings will be visible on the completed vessel, or a coded marking, acceptable to the Inspector, shall be used to assure identification of each piece of material during fabrication and subsequent identification of the markings on the completed vessel. In either case an as-built sketch or a tabulation of materials shall be made, identifying each piece of material with the certified test report or certificate of compliance and the coded marking. Except as indicated in paragraph 6.1.1.2.b, material may be marked by any method acceptable to the Inspector. The Inspector need not witness the transfer of the marks but shall satisfy himself that it has been correctly done.

b) Method of Transferring Markings – Where the service conditions prohibit die stamping for material identification, and when so specified by the user, the required data shall be marked on the plates in a manner that will allow positive identification upon delivery. The markings must be recorded so that each plate will be positively identified in its position in the finished vessel to the satisfaction of the Inspector. Transfer of markings for material that is to be divided shall be done in accordance with paragraph 6.1.1.2 a.

c) Transfer of Markings by Other than the Manufacturer – If the material is formed into shapes by anyone other than the Manufacturer of the completed pressure vessel and the original markings as required by the applicable material specification are unavoidably cut out, or the material is divided into two or more parts, then the manufacturer of the shape shall either:

   1) Transfer the original identification markings to another location on the shape; or

   2) Provide for identification by the use of a coded marking traceable to the original required marking, using a marking method agreed upon and described in the Quality Control System of the Manufacturer of the completed pressure vessel.

d) The material test report for this material, in conjunction with the above modified marking requirements, shall be considered sufficient to identify these shapes. Manufacturer's Partial Data Reports and parts stamping are not a requirement unless there has been fabrication to the shapes that includes welding beyond that exempted by paragraph 3.2.8.2.

6.1.1.3 Repair of Defective Materials

Defects may be removed and the material repaired by the vessel Manufacturer or, unless prohibited by the material specification, may also be repaired by the material manufacturer with the approval of the vessel Manufacturer. Material repairs that exceed those permitted by the material specification shall be made to the satisfaction of the Inspector. All repairs shall be in accordance with the following paragraphs.

a) Examination of Defective Areas – Areas from which defects have been removed shall be examined by either the magnetic particle or by the liquid penetrant method in accordance with Part 7 to ensure complete removal of the defect.

b) Repair by Welding – For the repair of materials by welding, the procedures and welds must be qualified in accordance with Section IX of the Code. If the base metal requires impact testing in accordance with paragraph 3.11, then a procedure test plate shall be welded and the deposited weld metal impact tested in accordance with paragraph 3.11 and shall meet the same minimum requirements as established for the base material. The repaired material shall be heat treated in accordance with the postweld heat treatment requirements of paragraph 6.4.2 when required.

c) Examination of Finished Weld-Repaired Surfaces

   1) The finished surface of the weld repair shall be prepared and inspected by either the magnetic particle or by the liquid penetrant method in accordance with Part 7.

   2) The area repaired by welding shall be examined by radiography in accordance with Part 7 if the depth of the weld deposit exceeds either 10 mm (3/8 in.) or one-half the material thickness.

d) All repairs to materials shall be documented (see paragraph 2.3.5).

6.1.2 Forming

6.1.2.1 Forming Shell Sections and Heads

All materials for shell sections and for heads shall be formed to the required shape by any process that will not unduly impair the mechanical properties of the material.

6.1.2.2 Thickness for Forming

The selected thickness of material shall be such that the forming processes will not reduce the thickness of the material at any point below the minimum value required by the design computation.

6.1.2.3 Forming of Carbon and Low Alloy Material Parts

a) Plates shall not be formed cold by blows.

b) Plates may be formed by blows at a forging temperature provided the plate is subsequently heat treated per the PWHT requirements of this Division.

c) Except when made of P-No. 1 Group Nos. 1 and 2 materials, all vessel shell sections, heads, and other pressure boundary parts fabricated by cold forming shall be heat treated subsequently (see paragraph 6.4.2) when the extreme fiber elongation (strain) exceeds 5% from the as-rolled condition. For P-No. 1 Group Nos. 1 and 2, this subsequent heat treatment is required if the extreme fiber elongation exceeds 40% or if the extreme fiber elongation exceeds 5% and any of the following conditions exist:

   1) The MDMT determined in accordance with paragraph 3.11 requires that the material be impact tested.

   2) The reduction by cold forming from the as-rolled thickness is more than 10% at any location where the extreme fiber elongation exceeds 5%.

   3) The temperature of the material during forming is in the range of 120°C to 480°C (250°F to 900°F).

   For all other P numbers and Group numbers, suitable heat treatment shall be applied to the formed part when the strain (extreme fiber elongation) exceeds 5% even though none of the conditions listed in paragraphs (1) through (3) exist.

d) The extreme fiber elongation shall be determined by the equations in Table 6.1.

e) When vessel shell sections, heads, or other pressure parts of carbon or low alloy steel plate are cold formed by other than the Manufacturer of the vessel, the required certification for the part shall indicate whether or not the part has been heat treated.

6.1.2.4 Forming High Alloy Material Parts

If the following conditions prevail, the cold formed areas of pressure-retaining parts manufactured of austenitic alloys shall be solution annealed by heating at the temperatures given in Table 6.2 for 0.8 min/mm (20 min/in.) of thickness or 10 minutes, whichever is greater, followed by rapid cooling.

a) The finishing-forming temperature is below the minimum heat-treating temperature given in Table 6.2; and

b) The design metal temperature and the forming strains exceed the limits shown in Table 6.2. The forming strains shall be calculated using the equations in Table 6.1. If the forming strains cannot be calculated as shown in this table, the Manufacturer shall determine the maximum forming strain.

c) For flares, swages, or upsets, heat treatment in accordance with Table 6.2 shall apply regardless of the amount of strain.

6.1.2.5 Forming Nonferrous Material Parts

a) A shell section that has been formed by rolling may be brought true-to-round for its entire length by pressing or rolling.

b) If the following conditions prevail, the cold formed areas of pressure-retaining parts manufactured of the alloys listed in Table 6.3 shall be solution annealed by heating at the temperatures given in Table 6.3 for 0.8 min/mm (20 min/in.) of thickness or 10 minutes, whichever is greater, followed by rapid cooling.

   1) The finishing-forming temperature is below the minimum heat-treating temperature given in Table 6.3; and

   2) The design metal temperature and the forming strains exceed the limits shown in Table 6.3. The Forming strains shall be calculated using the equations in Table 6.1. If the forming strains cannot be calculated as shown in this table, then the Manufacturer shall have the responsibility to determine the maximum forming strain.

c) For flares, swages, or upsets, heat treatment in accordance with Table 6.3 shall apply regardless of the amount of strain.

6.1.2.6 Preliminary Shaping Of Edges Of Plates To Be Rolled – If the plates are to be rolled, the adjoining edges of longitudinal joints of cylindrical vessels shall first be shaped to the proper curvature by preliminary rolling or forming in order to avoid having objectionable flat spots along the completed joints (see paragraph 6.1.2.7).

6.1.2.7 Forming Tolerances for vessel parts shall be in accordance with the following requirements.

a) Shells and Heads Subject to Internal Pressure – tolerances for shells and heads fabricated from plate that are subject to internal pressure are stipulated in paragraph 4.3.2.

b) Shells and Heads Subject to External Pressure – tolerances for shells and heads fabricated from plate that are subject to external pressure are stipulated in paragraph 4.4.4.

c) Tolerances for Shells Fabricated from Pipe – vessel shells fabricated from pipe, meeting all other requirements of this Part, may have variations of diameter permitted by the specification for such pipe.

d) Tolerances for Heads Fabricated from Pipe Caps – vessel heads fabricated from pipe caps, meeting all other requirements of this Part, may have variations of shape permitted by the applicable product specification.

6.1.2.8 Lugs and Fitting Attachments

All lugs, brackets, saddle type nozzles, manhole frames, reinforcement around openings, and other appurtenances shall be formed and fitted to conform to the curvature of the shell or surface to which they are attached. .

a) If pressure parts, such as saddle type nozzles, manhole frames, and reinforcement around openings, extend over pressure retaining welds, these welds shall be ground flush for the portion of the weld to be covered.

b) If nonpressure parts, such as lugs, brackets, and support legs and saddles, extend over pressure retaining welds, these welds shall be ground flush as described in subparagraph a) above, or such parts shall be notched or coped to clear those welds.

6.1.2.9 Spin-Holes

Spin-holes are permitted at the center of heads to facilitate forming. Spin-holes with a diameter less than or equal to 60 mm (2 3/8 in.) may be closed with a full-penetration weld using either a welded plug or weld metal. The thickness of the weld and plug shall be greater than or equal to the thickness of the head material adjacent to the spin-hole. The finished weld shall be examined by the magnetic particle or liquid penetrant method in accordance with Part 7. Full volumetric inspection of the weld in accordance with Part 7 shall be performed in addition to any examinations required by the material specification.

6.1.3 Base Metal Preparation

6.1.3.1 Examination of Materials

a) All materials to be used in constructing a pressure vessel shall be examined before fabrication for the purpose of detecting, as far as possible, defects that will affect the safety of the vessel. As fabrication progresses, the vessel Manufacturer shall carefully examine the edges of base materials (including the edges of openings cut through the thickness) to detect defects that have been uncovered during fabrication, and they shall be repaired in accordance with paragraph 6.1.1.3.

b) Except as required in paragraph 6.1.3.1.c, cut edges in base materials with thicknesses over 38 mm (1 1/2 in.) that are to be welded shall be examined for discontinuities by the magnetic particle or liquid penetrant method in accordance with Part 7.

c) For openings, cut edges in base materials in all thicknesses shall be examined for discontinuities as specified below using the magnetic particle method or liquid penetrant method in accordance with Part 7. Additional testing of materials prior to fabrication should be considered by the purchaser (i.e. ultrasonic testing of plate to SA-435 or SA-578, and of forgings to SA-388) for those services in which laminar discontinuities may be harmful.

   1) Examination is required for openings shown in Table 4.2.10, Details 1, 2, and 8.

   2) For other types of openings, this examination is not required for the cut edges of openings 75 mm (3 in.) in diameter or smaller.

   3) Non-laminar discontinuities (having length not parallel to the material surface) shall be removed.

   4) Discontinuities parallel to the surface, such as inclusions, which are disclosed by either method, are acceptable without repair if they do not exceed 25 mm (1 in.) in length.

d) If a pressure part is to be welded to a flat plate thicker than 13 mm (1/2 in.) to form a corner joint in accordance with paragraph 4.2.5.4.d, then the weld joint preparation and the peripheral edges of the flat plate forming a corner joint shall be examined as described below before welding by either the magnetic particle or liquid penetrant method in accordance with Part 7. After welding, the peripheral edge of the flat plate and any remaining exposed surface of the weld joint preparation shall be reexamined as specified below.

1) The weld edge preparation of typical joint preparations in flat plate as shown in Table 4.2.6, Details 1 through 3 and Table 4.2.8, Detail 1.

2) The outside peripheral edge of the flat plate after welding as shown in Table 4.2.6, Details 1 and 2.

3) The outside peripheral edge of the flat plate after welding as shown in Table 4.2.6, Details 3, if the distance from the edge of the completed weld to the peripheral edge of the flat plate is less than the thickness of the flat plate.

4) The inside peripheral surface of the flat plate after welding as shown in Table 4.2.9, Details 3 and 4.

6.1.3.2 Cutting Plates and Other Stock

a) Plates, edges of heads, and other parts may be cut to shape and size by mechanical means such as machining, shearing, and grinding; or by thermal cutting. After thermal cutting, all slag and detrimental discoloration of material, which has been molten shall be removed by mechanical means prior to further fabrication or use. When thermal cutting is used, the effect on mechanical properties shall be taken into consideration. The edges to be welded shall be uniform and smooth.

b) In general, nonferrous materials cannot be cut by the conventional oxyfuel equipment commonly used for steels. They may be melted and cut by oxyfuel powder cutting, carbon arc, oxygen arc, and other means. When such thermal means for cutting are employed, a shallow contaminated area adjacent to the cut results. This contamination shall be removed by grinding, machining, or other mechanical means after thermal cutting and prior to use or further fabrication.

6.1.3.3 Shearing of Nozzles and Manhole Necks

The ends of nozzles or manhole necks that are to remain unwelded in the completed vessel may be cut by shearing provided sufficient additional material is removed by any other method that produces a smooth finish. The cut edges shall be examined by the magnetic particle or liquid penetrant method in accordance with Part 7.

6.1.4 Fitting and Alignment

6.1.4.1 Parts that are being welded shall be fitted, aligned, and retained in position during the welding operation. If two parts are joined by the inertia and continuous drive friction welding processes, then one of the two parts must be held in a fixed position and the other part rotated. The two faces to be joined must be essentially symmetrical with respect to the axis of rotation. Some of the basic types of applicable joints are solid round to solid round, tube to tube, solid round to tube, solid round to plate, and tube to plate.

6.1.4.2 Means for Maintaining Alignment During Welding

Bars, jacks, clamps, tack welds, or other appropriate means may be used to maintain the alignment of the edges to be welded. Tack welds, if used to maintain alignment, shall either be removed completely when they have served their purpose, or their stopping and starting ends shall be properly prepared by grinding or other suitable means so that they may be satisfactorily incorporated into the final weld. Tack welds shall be made by qualified procedures and welders, shall be examined visually for defects, and, if found to be defective, shall be removed. If the work is done under the provisions of paragraph 2.3, then it is not necessary that a subcontractor making such tack welds for a vessel or parts manufacturer be a holder of a Code Certificate of Authorization. Temporary tack welds used to secure the shape of a component during handling or transportation are not required to be deposited by a holder of a Code Certificate of Authorization, provided they will be completely removed and will not be incorporated into the final weld. In addition, the areas shall be examined by the magnetic particle or liquid penetrant methods in accordance with Part 7.

6.1.4.3 Aligning Edges of Butt Joints

The edges of butt joints shall be held during welding so that the tolerances in paragraph 6.1.6 are not exceeded in the joint. When fitted girth joints have deviations exceeding the permitted tolerances, the head or shell ring, whichever is out-of-tolerance, shall be reformed until the allowable limits (see paragraph 6.1.6.1.a) are satisfied.

6.1.4.4 Removal of Temporary Attachments

The areas from which temporary attachments have been removed shall be dressed smooth and shall be examined by the magnetic particle method or liquid penetrant method in accordance with Part 7. Defects shall be removed and the material shall be inspected to ensure that the defects have been removed. If weld repairs are necessary, then the repairs shall be made using qualified welding procedures and welders, and the repairs shall be examined in accordance with paragraph 6.1.1.3.c.

6.1.5 Cleaning of Surfaces to Be Welded

6.1.5.1 The surfaces to be welded shall be clean and free of scale, rust, oil, grease, slag, detrimental oxides, and other deleterious foreign material. The method and extent of cleaning should be determined based on the material to be welded and the contaminants to be removed. When weld metal is to be deposited over a previously welded surface, all slag shall be removed by a roughing tool, chisel, chipping hammer, or other suitable means so as to prevent inclusion of impurities in the weld.

6.1.5.2 Cast surfaces to be welded shall be machined, chipped, or ground to remove foundry scale and to expose sound metal.

6.1.5.3 The requirements in paragraphs 6.1.5.1 and 6.1.5.2 above are not intended to apply to any process of welding by which proper fusion and penetration are otherwise obtained and by which the weld remains free from defects.

6.1.6 Alignment Tolerances for Edges to Be Butt Welded

6.1.6.1 Alignment of sections at edges to be butt welded shall be such that the maximum offset is not greater than shown below. Alternatively, offsets greater than those permitted below are allowable provided that the maximum offset is acceptable to the Inspector prior to welding and the requirements of paragraph 4.14 or Part 5, and the requirements of paragraph 6.1.6.2 are satisfied.

a) Cylindrical Shells – The maximum allowable offset in welded joints in cylindrical shells shall be as given in Table 6.4.

b) Spherical Shells and for Hemispherical Heads Welded to Cylindrical Shells – Joints in spherical vessels, joints within heads, and joints between cylindrical shells and hemispherical heads shall meet the requirements in Table 6.4.

c) Alignment tolerances at edges to be butt welded for quenched and tempered high strength steels shall be in accordance with paragraph 6.6.5.4.

6.1.6.2 Fairing of Offsets within Allowable Tolerances

Any offset within the allowable tolerance provided above shall be faired at a 3:1 taper over the width of the finished weld or, if necessary, by adding additional weld metal beyond what would have been the edge of the weld. If additional weld metal buildup is used, then it shall satisfy the requirements of paragraph 6.2.4.9.

6.1.6.3 Peaking Of Welds in Shells and Heads For Internal Pressure

a) If a fatigue analysis is required, see paragraph 4.1.1.4, then the peaking height, d_p, at Category A weld joints shall be measured by either an inside or outside template, as appropriate (see Figure 6.1). As an alternative, the peaking angle may be determined using the procedure described in Part 8 of API 579-1/ASME FFS-1 for materials operating outside the creep regime, or Part 10 of API 579-1/ASME FFS-1 for materials operating within the creep regime.

b) The chord length of the template shall be the larger of $D/6$ or 300 mm (12 in.), but need not exceed 900 mm (36 in.). If the weld joint is in a torispherical or an ellipsoidal head, the inside diameter, D, shall be equal to the diameter of the spherical portion of the head. If the weld joint is in a 2:1 ellipsoidal head, D shall be equal to 1.8 times the nominal inside diameter of the attached cylindrical shell.

c) If a fatigue analysis is required, the allowable value of d_p shall be determined using paragraph 4.14 and shall be shown in the Manufacturer's Design Report.

6.2 Welding Fabrication Requirements

6.2.1 Welding Processes

6.2.1.1 The welding processes that may be used in the construction of vessels under this Part are shown in Table 6.5. Definitions are given in Section IX that includes variations of these processes.

6.2.1.2 Welding of titanium is to be by gas tungsten arc welding (GTAW), gas metal arc welding (GMAW), plasma arc welding (PAW), electron beam welding (EBW), or laser beam welding (LBW), as defined in Section IX.

6.2.2 Welding Qualifications and Records

6.2.2.1 Manufacturer's Responsibility

a) Each Manufacturer or parts manufacturer is responsible for the welding done by his organization. The Manufacturer or parts manufacturer shall establish the procedure and be responsible for the tests required in Section IX with any additional tests required by this Division to qualify the welding procedures and the performance of welders and welding operators who apply these procedures.

b) Welders not in the employ of the Manufacturer may be used to fabricate pressure vessels constructed in accordance with this Division, provided all the following conditions are met.

   1) The Manufacturer shall be responsible for Code compliance of the completed pressure vessel or part, including Code symbol stamping and providing Data Report Forms properly executed and countersigned by the Inspector.

   2) All welding shall be performed in accordance with the Manufacturer's Welding Procedure Specifications, in accordance with the requirements of Section IX.

   3) All welders shall be qualified by the Manufacturer in accordance with the requirements of Section IX.

   4) The Manufacturer's Quality Control System shall include as a minimum:

      i) A requirement for complete and exclusive administrative and technical supervision of all welders by the Manufacturer

      ii) Evidence of the Manufacturer's authority to assign and remove welders at his discretion without involvement of any other organization

      iii) A requirement for assignment of welder identification symbols

      iv) Evidence that this program has been accepted by the Manufacturer's Authorized Inspection Agency.

6.2.2.2 Qualification Test Limitations

Welding of all test coupons shall be conducted by the Manufacturer. Testing of all test coupons shall be the responsibility of the Manufacturer. Alternatively, the AWS Standard Welding Procedure Specifications that have been accepted by Section IX may be used, provided these specifications meet all other requirements of this Division. Qualification of a welding procedure by one Manufacturer shall not qualify that procedure for use by any other Manufacturer, except as provided for in QW-201 of Section IX. A performance qualification

test conducted by one Manufacturer shall not qualify a welder or welding operator to do work for any other Manufacturer, except as provided for in QW-300 of Section IX.

6.2.2.3 Production Welding Prior to Qualification

Production welding shall not be undertaken until after the welding procedures that are to be used have been qualified.

6.2.2.4 Qualification of Welding Procedure

a) Each welding procedure that is to be followed in construction shall be recorded in detail by the Manufacturer.

b) The procedure used in welding pressure parts and in joining load-carrying nonpressure parts, such as attachments, to pressure parts shall be qualified in accordance with Section IX.

c) When making procedure test plates for butt welds in accordance with Section IX, consideration shall be given to the effect of angular, lateral, and end restraint on the weldment. This applies particularly to material and weld metal with an ultimate tensile strength of 550 MPa (80,000 psi) or higher and thick sections of both low and high tensile strength material. The addition of restraint during the welding may result in cracking that otherwise might not occur.

d) The procedure used in welding nonpressure-bearing attachments, which have essentially no load-carrying function (such as extended heat transfer surfaces, insulation support pins, etc.), to pressure parts shall meet the following requirements.

    1) If the welding process is manual, machine, or semiautomatic, then the procedure qualification is required in accordance with Section IX.

    2) If the welding is performed using any automatic welding process performed in accordance with a Welding procedure Specification (in compliance with Section IX as far as applicable), then the procedure qualification testing is not required.

6.2.2.5 Tests of Welders and Welding Operators

a) Welders and the welding operators used in welding pressure parts and in joining load-carrying nonpressure parts (attachments) to pressure parts shall be qualified in accordance with Section IX.

    1) The qualification test for welding operators of machine welding equipment shall be performed on a separate test plate prior to the start of welding or on the first work piece.

    2) When stud welding is used to attach load-carrying studs, a production stud weld test of the procedure and welding operator shall be performed on a separate test plate or tube prior to the start of welding on each work shift. This weld test shall consist of five studs, welded and tested in accordance with either the bend or torque stud weld testing described in Section IX.

b) Welders and welding operators used in welding nonpressure-bearing attachments, that have essentially no load-carrying function (such as extended heat transfer surfaces, insulation support pins, etc.), to pressure parts shall comply with the following.

    1) If the welding process is manual, machine, or semiautomatic, qualification in accordance with Section IX is required.

    2) If welding is done by any automatic welding process, performance qualification testing is not required.

    3) If stud welding is used, a production stud weld test, appropriate to the end use application requirements, shall be specified by the Manufacturer and carried out on a separate test plate or tube at the start of each shift.

6.2.2.6 Maintenance of Qualification Records

The Manufacturer shall maintain a record of the welding procedures and the welders and welding operators employed by the Manufacturer. The record shall indicate the date and results of tests and the identification mark assigned to each welder. These records shall be maintained in accordance with Section IX.

6.2.3 Precautions to Be Taken Before Welding

6.2.3.1 Identification, Handling, and Storing of Electrodes and Other Welding Materials

The Manufacturer is responsible for control of the welding electrodes and other materials which are to be used in the fabrication of the vessel. Suitable identification, storage, and handling of electrodes, flux, and other welding materials shall be maintained. Precautions shall be taken to minimize absorption of moisture by electrodes and flux. The methods used shall be documented in the Manufacturer's Quality Assurance Manual.

6.2.3.2 Lowest Permissible Minimum Temperature for Welding

No welding of any kind shall be done when the temperature of the metal is lower than -20°C (0°F). If the temperature is between 0°C (32°F) and -20°C (0°F), then the surface of all areas within 75 mm (3 in.) of the point where a weld is to be started shall be heated to a temperature at least warm to the hand (estimated to be above 15°C (60°F)) before welding is started. No welding shall be done when surfaces are wet or covered with ice, when snow is falling on the surfaces to be welded, or during periods of high wind unless the welders or welding operators and the work are properly protected.

6.2.4 Specific Requirements for Welded Joints

6.2.4.1 Type No.1 Butt Joints

a) Definition – Type No. 1 butt joints are defined in paragraph 4.2.5.1.a and Table 4.2.2.

b) Weld Penetration and Reinforcement

    1) Butt welded joints shall have complete penetration and full fusion. As-welded surfaces are permitted; however, the surface of the welds shall be sufficiently free from coarse ripples, grooves, overlaps, and abrupt ridges and valleys to permit proper interpretation of radiographic and other required nondestructive examinations. If it is suspected that an indication on a radiograph is due to the surface condition of the weld, the radiograph shall be compared to the actual weld surface to aid in interpretation.

    2) A reduction in thickness due to the welding process is acceptable provided all of the following conditions are met. Note that it is not the intent of this paragraph to require measurement of reductions in thickness due to the welding process. If a disagreement between the Manufacturer and the Inspector exists as to the acceptability of any reduction in thickness, the depth shall be verified by actual measurement.

        i) The reduction in thickness shall not reduce the material of the adjoining surfaces below the minimum required thickness at any point.

        ii) The reduction in thickness shall not exceed 0.8 mm (1/32 in.) or 10% of the nominal thickness of the adjoining surface, whichever is less.

c) Examination Requirements – Examination requirements shall be in accordance with Part 7.

d) Weld Reinforcement – To assure that the weld grooves are completely filled so that the surface of the weld metal at any point does not fall below the surface of the adjoining base materials, weld metal may be added as reinforcement on each face of the weld. The thickness of the weld reinforcement on each face shall not exceed that shown in Table 6.6. Concavity due to the welding process on the root side of a single welded circumferential butt weld is permitted when the resulting thickness of the weld is at least equal to the thickness of the thinner member of the two sections being joined and the contour of the concavity is smooth.

6.2.4.2 Type No.2 Butt Joints

a) Definitions – Type No.2 butt joints are defined in paragraph 4.2.5.1.a and Table 4.2.2.

b) Penetration and Reinforcement – If Type No.2 butt joints are used, then particular care shall be taken in aligning and separating the components to be joined so that there will be complete penetration and fusion at the bottom of the joints for their full length. However, for assuring complete filling of the weld

grooves, weld reinforcement in accordance with the limits specified in paragraph 6.2.4.1.d need be supplied only on the side opposite the backing strip.

c) Backing Strips – Backing strips shall be continuous and any splices shall be butt welded. Circumferential single-welded butt joints with one plate offset to form a backing strip are prohibited.

d) Examination Requirements – Examination requirements shall be in accordance with Part 7.

6.2.4.3 Full Penetration Corner Joints

a) Definition – Full penetration corner joints are joints defined in paragraph 4.2.5.1.c.

b) Penetration and Fusion – Welds in full penetration comer joints shall be groove welds extending completely through at least one of the parts being joined and shall be fully fused to each part.

c) Examination Requirements – Examination requirements shall be in accordance with Part 7.

6.2.4.4 Partial Penetration Corner Joints for Nozzle Attachments

a) Definition – Partial penetration corner joints are defined in paragraph 4.2.5.5.c.8 and Table 4.2.14.

b) Penetration Requirements – Partial penetration welds shall have a minimum depth of penetration equal to that required by Table 4.2.14.

c) Examination Requirements – Examination requirements shall be in accordance with Part 7.

6.2.4.5 Fillet Welded Joints

a) Definition – Fillet welded joints are defined in paragraph 4.2.5.1.d and Table 4.2.2.

b) Quality Requirements – The weld metal for a fillet weld shall be deposited in such a way that adequate penetration into the base metal at the root of the weld is secured. The reduction of the thickness of the base metal due to the welding process at the edges of the fillet weld shall meet the same requirements as for butt welds. (see paragraph 6.2.4.1.b)

c) Examination Requirements – Examination requirements shall be in accordance with Part 7.

6.2.4.6 Welds Attaching Nozzles and Other Connections

The design requirements for welds attaching nozzle necks and other connections are given in paragraph 4.2.5.5.

6.2.4.7 Welds Attaching Nonpressure Parts and Stiffeners

a) The design requirements for welds attaching nonpressure parts and stiffeners to pressure parts are given in paragraph 4.2.5.6.

b) Examination Requirements – Examination requirements shall be in accordance with Part 7.

6.2.4.8 Austenitic Chromium-Nickel Alloy Steel Welds

All austenitic chromium-nickel alloy steel welds, both butt and fillet, in parts with a shell thickness that exceeds 19 mm (0.75 in.) shall be examined by the liquid penetrant method in accordance with Part 7. This examination shall be made following heat treatment, if heat treatment is performed. All cracks shall be repaired.

6.2.4.9 Surface Weld Metal Buildup

a) Construction in which deposits of weld metal are applied to the surface of base metal for the purpose of restoring the thickness of the base metal or modifying the configuration of weld joints in order to provide the tapered transition requirements of paragraph 4.2.

b) Procedure Qualification – A butt welding procedure qualification in accordance with the provisions of Section IX shall be performed for the thickness of weld metal deposited, prior to production welding.

c) Examination Requirements

   1) All weld metal build-up shall be examined over the full surface of the deposit by the magnetic

particle or liquid penetrant method in accordance with Part 7. This requirement does not apply to weld overlay.

2) When such surface weld metal buildup is used in welded joints that require radiographic examination in accordance with Part 7, the weld metal buildup shall be included in this examination.

6.2.5 Miscellaneous Welding Requirements

6.2.5.1 Preparation of Reverse Side of Double-Welded Joints

The reverse side of double-welded joints shall be prepared by chipping, grinding, or melting out, so as to secure sound metal at the base of the weld metal first deposited before applying weld metal from the reverse side. These requirements are not intended to apply to any process of welding by which proper fusion and penetration are otherwise obtained and by which the base of the weld remains free from impurities.

6.2.5.2 Aligning and Separating Components of Single-Welded Joints

If single-welded joints are used, the components to be joined shall be aligned and separated so that there will be complete penetration and fusion at the bottom of the joint for its full length.

6.2.5.3 Peening

a) Weld metal and heat affected zones may be peened by manual, electric, or pneumatic means when it is deemed necessary or helpful to control distortion, to relieve residual stresses, improve fatigue life, or to improve the quality of the weld. Peening shall not be used on the initial (root) layer of weld metal or on the final (face) layer unless the weld is subsequently postweld heat treated. In no case, however, is peening to be performed in lieu of any postweld heat treatment required by these rules.

b) Controlled shot peening and other similar methods, which are intended only to enhance surface properties of the vessel or vessel parts, shall be performed after any nondestructive examinations and pressure tests required by these rules.

6.2.5.4 Identification Markings or Records for Welders and Welding Operators

a) Each welder and welding operator shall stamp the identifying number, letter, or symbol, assigned by the Manufacturer on or adjacent to and at intervals of not more than 0.9 m (3 ft.) along the welds they make in plates 6 mm (0.25 in.) and over in thickness. Alternatively, a record shall be kept by the Manufacturer of those employed on welding each joint that shall be made available to the Inspector.

b) If a multiple number of permanent nonpressure part load bearing attachment welds, non-load-bearing welds, such as stud welds, or special welds, such as tube-to-tubesheet welds, are made on a vessel, the Manufacturer need not identify the welder or welding operator that welded each individual joint provided:

   1) The Manufacturer's Quality Control System includes a procedure that will identify the welders or welding operators that made such welds on each vessel so that the Inspector can verify that the welders or welding operators were all properly qualified; and

   2) The welds in each category are all of the same type and configuration and are welded with the same Welding Procedure Specification.

c) Permanent identification of welders or welding operators making tack welds that become part of the final pressure weld is not required, provided the Manufacturer's Quality Control System includes a procedure to permit the Inspector to verify that such tack welds were made by qualified welders or welding operators.

6.2.5.5 Friction Welding Visual Examination

The welded joint between two members joined by the inertia and continuous drive friction welding processes shall be a full penetration weld. Visual examination of the as-welded flash roll of each weld shall be made as an in-process check. The weld upset shall meet the specified amount within ±10%. The flash shall be removed to sound metal.

6.2.5.6 Capacitor Discharge Welding

Capacitor discharge welding may be used for welding temporary attachments and permanent nonstructural attachments without postweld heat treatment provided the following requirements are met.

a) The Welding Procedure Specification shall be prepared in accordance with Section IX, insofar as possible describing the capacitor discharge equipment, the combination of materials to be joined, and the technique of application. Qualification of the Welding Procedure is not required.

b) The energy output shall be limited to 125 W-sec.

6.2.5.7 Burr Grinding of Completed Weld Joints

Burr grinding of weld joints to improve fatigue life performance shall be in accordance with Figure 6.2 when specified in the User's Design Specification. The remaining ligament after burr grinding (i.e. t – g, see Figure 6.2) shall be greater than or equal to the minimum required wall thickness for the component obtained using Part 4 or Part 5, as applicable.

6.2.5.8 Corrosion Resistance of Alloy Welds

Alloy welds that are exposed to the corrosive action or environmental degradation of the contents of the vessel should have a resistance to corrosion that is not substantially less than that of the base metal. The use of filler metal that will deposit weld metal with practically the same composition as the material joined is recommended. Alternatively, filler metal of a different composition may be used provided the strength of the weld metal at the operating temperature is not appreciably less than that of the high alloy material to be welded, and the user or an agent acting on behalf of the user is satisfied that its resistance to corrosion is satisfactory for the intended service. The columbium content of weld metal shall not exceed 1.00%, except that ENiCrMo-3, ERNiCrMo-3, and ENiCrMo-12 weld filler metal made to SFA-5.11 and SFA-5.14 may be used to weld S31254, S31603, S31703, S31725, and S31726 to a maximum design temperature of 480°C (900°F).

6.2.6 Summary of Joints Permitted and Their Examination

6.2.6.1 Types of Joints Permitted

The types of weld joints permitted for each Weld Category are given in paragraph 4.2.

6.2.6.2 Examination Requirements

Examination requirements shall be in accordance with Part 7.

6.2.7 Repair of Weld Defects

6.2.7.1 Removal of Unacceptable Defects

Unacceptable defects detected visually or by the examinations in Part 7, and defects detected by leakage tests, shall be removed by mechanical means or by thermal gouging processes.

6.2.7.2 Re-welding of Areas to Be Repaired

The areas to be repaired shall be re-welded by qualified welders using qualified welding procedures.

6.2.7.3 Examination of Repaired Welds

Repaired welds shall be reexamined by the methods of the original examination of the weld. The repaired weld shall not be accepted unless the examination shows the repair to be satisfactory.

6.2.7.4 Postweld Heat Treatment of Repaired Welds

The postweld heat treatment rules in paragraph 6.4 shall apply to all weld repairs.

6.2.8 Special Requirements for Welding Test Plates for Titanium Materials

6.2.8.1 If a vessel of welded titanium contains Category A or B weld joints, then a production test plate of the same specification, grade, and thickness shall be made of sufficient size to provide at least one face and one root bend specimen or two-side bend specimens dependent upon plate thickness. Where longitudinal joints are involved, the test plate shall be attached to one end of the longitudinal joint and welded continuously with the joint. Where circumferential joints only are involved, the test plate need not be attached but shall be welded along with the joint, and each welder or welding operator shall deposit weld metal in the test plate at the location and proportional to that deposited in the production weld.

6.2.8.2 Test plates shall represent each welding process or combination of processes or a change from machine to manual or vice versa. At least one test plate is required for each vessel, provided not over 30 m (100 ft.) of Category A or B joints are involved. An additional test plate, meeting the same requirements as outlined above, shall be made for each additional 30 m (100 ft.) of Category A or B joints involved. The bend specimens shall be prepared and tested in accordance with Section IX, QW-160. Failure of either bend specimen constitutes rejection of the weld.

6.3 Special Requirements for Tube-To-Tubesheet Welds

6.3.1 Material Requirements

Tubes may be attached to tubesheets by welding provided the tubes and tubesheets or tubesheet facings are of weldable materials covered by this Division.

6.3.2 Holes in Tubesheets

6.3.2.1 Preparing Holes in Tubesheets

Tube holes in tubesheets shall be drilled full size or they may be punched to three-quarters size and then drilled, reamed, or finished full size with a rotating cutter.

6.3.2.2 Clearance between Tubes and Tube Holes

The clearance between the outside surface of the tubes and the inside surfaces of the tube holes shall not exceed the clearance used in the welding procedure qualification tests.

6.3.2.3 Finish of Holes

The edges of the tubesheet at the tube holes on the side to be welded shall be free of burrs, and the edges of the tubesheet at the tube hole on the side opposite the weld shall have sharp corners removed. The surfaces of tube holes in tubesheets shall have a workmanship-like finish.

6.3.3 Weld Design and Joint Preparation

The weld dimensions and weld detail, and joint preparation, if used, shall comply with the details included in the Welding Procedure Specification.

6.3.4 Qualification of Welding Procedure

Tube-to-tubesheet welding procedure specifications shall be qualified in accordance with the requirements of QW-193 of Section IX.

6.4 Preheating and Heat Treatment of Weldments

6.4.1 Requirements for Preheating of Welds

6.4.1.1 The Welding Procedure Specification for the material being welded shall specify the minimum preheating requirements in accordance with the weld procedure qualification requirements of Section IX.

Where preheating is not required by the welding procedure, preheating may be employed during welding to assist in completion of the welded joint. The need for and temperature of preheat are dependent on a number of factors, such as the chemical analysis, degree of restraint of the parts being joined, elevated temperature physical properties, and material thicknesses.

6.4.1.2 Guidelines for preheating are provided in Table 6.7 for the materials listed by P-Numbers of Section IX. It is cautioned that the preheating parameters shown in this table do not necessarily ensure satisfactory completion of the welded joint, and requirements for individual materials for the P-Number listing may have preheating requirements that are more restrictive.

6.4.2 Requirements for Postweld Heat Treatment

6.4.2.1 Before applying the detailed requirements and exemptions in these paragraphs, satisfactory qualification of the welding procedures to be used shall be performed in accordance with all the variables of Section IX and paragraph 6.3.4, including conditions of postweld heat treatment or its omission, and the restrictions listed in this paragraph.

6.4.2.2 Postweld heat treatment requirements for all materials of construction are shown below.

a) Postweld heat treatment requirements for quenched and tempered high strength steel materials listed in Table 3.A.4, are covered in paragraph 6.6.6.

b) Postweld heat treatment requirements for nonferrous materials are covered in paragraph 6.4.6.

c) 2¼Cr-1Mo-¼V and 3Cr-1Mo-¼V-B Materials – The final postweld heat treatment shall be in accordance with the requirements of this Division for P-No. 5C materials.

d) 2¼Cr-1Mo Materials – The final postweld heat treatment temperature shall be in accordance with the requirements of this Division for P-No. 5A materials, except that, for the materials listed in Table 3.1, the permissible minimum normal holding temperature is 650 °C (1200°F) and the holding time shall be 2.5 minutes/mm (1 hr/in.). For thicknesses over 125 mm (5 in.), the holding time shall be 5 hours plus 0.6 min for each additional mm (15 minutes for each additional inch) over 125 mm (5 in.).

e) Postweld heat treatment requirements for all other materials are covered in the tables listed below. Except as otherwise provided in these tables, all welds in pressure vessels or pressure vessel parts shall be given a postweld heat treatment at a temperature not less than that specified in the applicable table based on the nominal thickness as defined in paragraph 6.4.2.7. The materials shown below are identified by P-Numbers and Group Numbers in accordance with QW /QB-422 of Section IX and are provided in Annex 3.A. for each material specification.

    1) Table 6.8 – Materials P-No. 1, Group 1, 2, 3
    2) Table 6.9 – Materials P-No. 3, Group 1, 2, 3
    3) Table 6.10 – Materials P-No. 4, Group 1, 2.
    4) Table 6.11 – Materials P-No. 5A, P-No. 5B Group 1, and P-No. 5C Group 1, P-No. 5B Group 2
    5) Table 6.12 – Materials P-No. 6, Group 1, 2, 3
    6) Table 6.13 – Materials P-No. 7, Group 1, 2 and P-No. 8
    7) Table 6.14 – Materials: P-No. 9A, Group 1 and P-No. 9B, Group 1
    8) Table 6.15 – Materials: P-No. 10A, Group 1; P-No. 10B, Group 2; P-No. 10C, Group 1, P-No. 10E, Group 1; P-No. 10F, Group 6; P-No. 10G, Group 1; P-No. 10H, Group 1; P-No. 10I, Group 1; and P-No. 10K, Group 1
    9) Table 6.16 – Alternative Postweld Heat Treatment Requirements For Carbon And Low Alloy Steels

6.4.2.3 Additional postweld heat treatment requirements, other than those cited in paragraph 6.4.2.2.e, may be mandatory based on the requirements of paragraph 3.11.

6.4.2.4 The exemptions from postweld heat treatment permitted in paragraph 6.4.2.2 are not permitted when welding ferritic materials greater than 3 mm (1/8 in.) thick with the electron beam welding process, or

when welding P-No 3, P-No 4, P-No 5A, P-No 5B, P-No 5C, P-No 6, P-No 7 (except for Type 405 and Type 410S), and P-No 10 materials using the inertia and continuous drive friction welding process.

6.4.2.5 When Holding Temperatures and Times May Be Exceeded

Except where prohibited in paragraph 6.4.2.2, holding temperatures and/or holding times in excess of the minimum values given in these tables may be used. A time-temperature recording of all postweld heat treatments shall be provided for review by the Inspector. The holding time at temperature specified in the table references of paragraph 6.4.2.2 need not be continuous. It may be an accumulation of time of multiple postweld heat treat cycles.

6.4.2.6 Heat Treatment of Pressure Parts Consisting of Different P-Number Groups

When pressure parts of two different P-Number groups are joined by welding, the postweld heat treatment shall be that required by the material having the higher postweld heat treatment temperature. When nonpressure parts are welded to pressure parts, the postweld heat treatment temperature of the pressure part shall control.

6.4.2.7 Definition of Nominal Thickness Governing Postweld Heat Treatment

The nominal thickness as used in this paragraph is the thickness of the welded joint as defined below. For pressure vessels or parts of pressure vessels being postweld heat treated in a furnace charge, it is the greatest weld thickness in any vessel or vessel part which has not previously been postweld heat treated.

a) When the welded joint connects parts of the same thickness, using a full penetration butt weld, the nominal thickness is the total depth of the weld exclusive of any permitted weld reinforcement.

b) For groove welds, the nominal thickness is the depth of the groove.

c) For fillet welds, the nominal thickness is the throat dimension. If a fillet weld is used in conjunction with a groove weld, the nominal thickness is the depth of the groove or the throat dimension, whichever is greater.

d) For stud welds, the nominal thickness shall be the diameter of the stud.

e) When a welded joint connects parts of unequal thicknesses, the nominal thickness shall be the following:

   1) The thinner of two adjacent butt welded parts including head to shell connections;

   2) The thickness of the shell in connections to tubesheets, flat heads, covers, flanges, or similar constructions;

   3) For nozzles, the thickness of the weld across the nozzle neck or shell or head or reinforcing pad or attachment fillet weld, whichever is the greater;

   4) The thickness of the nozzle neck at the joint in nozzle neck to flange connections;

   5) The thickness of the weld at the point of attachment when a nonpressure part is welded to a pressure part;

   6) The thickness of the weld in tube-to-tubesheet connections.

f) For repairs, the nominal thickness is the depth of the repair weld.

6.4.2.8 Heat Treatment of Electroslag Welds in Ferritic Materials

Electroslag welds in ferritic materials over 38 mm (1 1/2 in.) in thickness at the joint shall be given a grain refining (austenitizing) heat treatment. For P-No.1 materials only, the heating and cooling rate restrictions of paragraphs 6.4.4.b and 6.4.4.e do not apply when the heat treatment following welding is in the austenitizing range.

6.4.3 Procedures for Postweld Heat Treatment

6.4.3.1 The postweld heat treatment shall be performed in accordance with one of the procedures of paragraph 6.4.3.1 through 6.4.3.7. In the procedures that follow, the soak band is defined as the volume of metal required to meet or exceed the minimum PWHT temperatures required in paragraph 6.4.2.2. As a minimum, the soak band shall contain the weld, heat affected zone, and a portion of base metal adjacent to

the weld being heat treated. The minimum width of this volume is the widest width of weld plus the nominal thickness defined in paragraph 6.4.2.7 or 50 mm (2 in.), whichever is less, on each side or end of the weld. For additional detail recommendations regarding implementation and performance of these procedures refer to Welding Research Council (WRC) Bulletin 452, June 2000, “Recommended Practices for Local Heating of Welds in Pressure Vessels”.

6.4.3.2 Heating Vessel in a Furnace in One Heat

Postweld heat treatment is performed by heating the vessel as a whole in a closed furnace. This procedure is preferable and should be used whenever practicable.

6.4.3.3 Heating Vessel Portions in a Furnace in More Than One Heat

Postweld heat treatment is performed by heating the vessel in more than one heat in a furnace, provided the overlap of the heated sections of the vessel is at least 1.5 m (5 ft.). When this procedure is used, the portion outside of the furnace shall be shielded so that the temperature gradient is not harmful. The cross section where the vessel projects from the furnace shall not intersect a nozzle or other structural discontinuity.

6.4.3.4 Heating Shell Sections, Heads, and Other Portions before Joining

Postweld heat treatment is performed by heating of shell sections, heads, and/or portions of vessels for postweld heat treatment of longitudinal joints or complicated welded details before joining to make the completed vessel. If it is not practical to postweld heat treat the complete vessel as a whole or in two or more heats as provided in paragraph 6.4.3.3, then any circumferential joints not previously postweld heat treated may be locally postweld heat treated by any appropriate means that will assure the required uniformity. For such local heating, the soak band shall extend around the full circumference. The portion outside the soak band shall be protected so that the temperature gradient is not harmful. This procedure may also be used to postweld heat treat portions of new vessels after repairs subject to the owner/user’s approval.

6.4.3.5 Heating Vessel Internally

Postweld heat treatment is performed by internally heating the vessel. The internal heating may be provided by any appropriate means. Adequate indicating and recording temperature devices shall be utilized to aid in the control and maintenance of a uniform distribution of temperature in the vessel wall. Previous to this operation, the vessel shall be fully enclosed with insulating material.

6.4.3.6 Local Heating of Nozzles to Vessels and External Attachments

a) Local heating of nozzles and attachments shall be performed by heating a circumferential band containing nozzles or other welded attachments that require postweld heat treatment in such a manner that the entire band shall be brought up uniformly to the required temperature and held for the specified time. Except as modified in the paragraph below, the soak band shall extend around the entire vessel and shall include the nozzle or welded attachment.
    1) The portion of the vessel outside of the circumferential soak band shall be protected so that the temperature gradient is not harmful; this procedure may also be used for local heat treatment of circumferential joints in pipe, tubing, or nozzle necks. In the latter case, proximity to the shell increases thermal restraint, and the designer should provide adequate length to permit heat treatment without harmful gradients at the nozzle attachment, or heat a full circumferential band around the shell, including the nozzle.
    2) The circumferential soak band width may be varied away from the nozzle or attachment weld requiring PWHT, provided the required soak band around the nozzle or attachment weld is heated to the required temperature and held for the required time. As an alternative to varying the soak band width, the temperature within the circumferential band away from the nozzle or attachment may be varied and need not reach the required temperature, provided the required soak band around the nozzle or attachment weld is heated to the required temperature, held for the required time, and the temperature gradient is not harmful throughout the heating and cooling cycle. The portion of the vessel outside of the circumferential soak band shall be protected so that the temperature gradient is not harmful.

b) The procedure in paragraph 6.4.3.6.a may also be used to postweld heat treat portions of vessels after repairs.

6.4.3.7 Local Area Heating of Double Curvature Heads or Shells

Postweld heat treatment is performed by heating a local area around nozzles or welded attachments in the larger radius sections of a double curvature head or a spherical shell or head in such a manner that the area is brought up uniformly to the required temperature and held for the specified time. The soak band shall include the nozzle or welded attachment. The minimum soak band size shall be a circle whose radius is the widest width of the weld attaching the nozzle, reinforcing plate, or structural attachment to the shell, plus the nominal thickness as defined in paragraph 6.4.2.7, or 50 mm (2 in.), whichever is less. The portion of the vessel outside of the soak band shall be protected so that the temperature gradient is not harmful.

6.4.3.8 Heating of Other Configurations

Postweld heat treatment performed by local area heating of other configurations such as “spot” or bulls eye” local heating not addressed in paragraphs 6.4.3.2 through 6.4.3.7 is permitted provided that other measures (based upon sufficiently similar documented experience or evaluation) are taken that consider the effect of thermal gradients, all significant structural discontinuities (such as nozzles, attachments, head-to-shell junctions), and any mechanical loads which may be present during PWHT. The portion of the vessel outside of the soak band shall be protected so that the temperature gradient is not harmful. The soak band shall include a circle that extends beyond the edges of the attachment weld in all directions by a minimum of the nominal thickness defined in paragraph 6.4.2.7 or 50 mm (2 in.), whichever is less.

6.4.4 Operation of Postweld Heat Treatment

The operation of postweld heat treatment shall be carried out by one of the procedures given in paragraph 6.4.3 in accordance with the following requirements.

a) When post weld heat treatment is performed in a furnace (see paragraph 6.4.3.2), the temperature of the furnace shall not exceed 430°C (800°F) at the time the vessel or part is placed in it.

b) Above 430°C (800°F), the rate of heating shall be not more than 220°C/hr (400°F /hr) divided by the maximum metal thickness of the shell or head plate in inches, but in no case more than 220°C/hr (400°F /hr) and in no case need it be less than 56°C/hr (100°F/hr). During the heating period there shall not be a greater variation in temperature throughout the portion of the vessel being heated than 140°C (250°F) within any 4.6 m (15 ft) interval of length.

c) The vessel or vessel part shall be held at or above the temperature specified in paragraph 6.4.2 for the period of time specified in this paragraph. During the holding period, there shall not be a difference greater than 85°C (150°F) between the highest and lowest temperatures throughout the portion of the vessel being heated, except where the range is further limited in paragraph 6.4.2.

d) When post weld heat treatment is performed in a furnace (see paragraph 6.4.3.2), during the heating and holding periods, the furnace atmosphere shall be so controlled as to avoid excessive oxidation of the surface of the vessel. The furnace shall be of such design as to prevent direct impingement of the flame on the vessel.

e) Above 430°C (800°F), cooling shall be done at a rate not greater than 280°C/hr (500°F/hr) divided by the maximum metal thickness of the shell or head plate in inches, but in no case need it be less than 55°C/hr (100°F/hr). From 430°C (800°F), the vessel may be cooled in still air.

6.4.5 Postweld Heat Treatment after Repairs

6.4.5.1 Except as permitted in paragraph 6.4.5.2 below, vessels or parts of vessels that have been postweld heat treated in accordance with the requirements of paragraph 6.4 shall again be postweld heat treated after welded repairs have been made.

6.4.5.2 Weld Repairs Made after Postweld Heat Treatment

Weld repairs to P-No.1 Gr. Nos. 1-3 materials and to P-No. 3 Gr. Nos. 1-3 materials, and to the weld metals used to join these materials, may be made after the final PWHT but prior to the final hydrostatic test without

additional PWHT provided that all the requirements below are satisfied. The welded repairs shall meet the requirements shown below. These requirements do not apply when the welded repairs are minor restorations of the material surface such as those required after the removal of construction fixtures and provided that the surface is not exposed to the vessel contents.

a) PWHT is not a service requirement defined by the user.

b) The material is not required to be impact tested to quality toughness properties in accordance with paragraph 3.11.

c) The Manufacturer shall give prior notification of the repair to the user or to his designated agent and shall not proceed until acceptance has been obtained. Such repairs shall be recorded on the Data Report.

d) The total repair depth shall not exceed 38 mm (1 1/2 in.) for P-No. 1 Gr. Nos. 1-3 materials and 16 mm (5/8 in.) for P-No. 3 Gr. Nos. 1-3 materials. The total depth of a weld repair shall be taken as the sum of the depths for repairs made from both sides of a weld at a given location.

e) After removal of the defect, the groove shall be examined, using either the magnetic particle or liquid penetrant examination methods in accordance with Part 7.

f) In addition to the requirements of Section IX for qualification of Welding Procedure Specifications for groove welds, the following requirements shall apply.

    1) The weld shall be deposited by the manual shielded metal arc process using low hydrogen electrodes. The electrodes shall be properly conditioned in accordance with Section II, Part C, SFA-5.5. The maximum bead width shall be four times the electrode core diameter.

    2) For P-No. 1 Gr. Nos. 1-3 materials, the repair area shall be preheated and maintained at a minimum temperature of 95°C (200°F) during welding.

    3) For P-No. 3 Gr. Nos. 1-3 materials, the repair weld method shall be limited to the SMAW half bead weld repair and weld temper bead reinforcement technique. The repair area shall be preheated and maintained at a minimum temperature of 175°C (350°F) during welding. The maximum interpass temperature shall be 230°C (450°F). The initial layer of weld metal shall be deposited over the entire area using 3 mm (1/8 in.) maximum diameter electrodes. Approximately one-half the thickness of this layer shall be removed by grinding before depositing subsequent layers. The subsequent weld layers shall be deposited using 4 mm (5/32 in.) maximum diameter electrodes in such a manner as to ensure tempering of the prior weld beads and their heat affected zones. A final temper bead weld shall be applied to a level above the surface being repaired without contacting the base material but close enough to the edge of the underlying weld bead to ensure tempering of the base material heat affected zone. After completing all welding, the repair area shall be maintained at a temperature of 205°C (400°F) to 260°C (500°F) for a minimum of 4 hours. The final temper bead reinforcement layer shall be removed substantially flush with the surface of the base material.

g) After the finished repair weld has reached ambient temperature, it shall be examined using the magnetic particle or liquid penetrant examination methods in accordance with Part 7. If the examination is by magnetic particle method, only the alternating current yoke type is acceptable. For P-No. 3 Gr. No.3 materials, the examination shall be made after the material has been at ambient temperature for a minimum period of 48 hours to determine the presence of possible delayed cracking of the weld. In addition, welded repairs greater than 10 mm (3/8 in.) deep in materials, and in welds that are required to be examined using the radiographic method in accordance with Part 7, shall be examined using the radiographic method in accordance with Part 7.

h) The vessel shall be hydrostatically tested after making the welded repair.

6.4.6 Postweld Heat Treatment of Nonferrous Materials

6.4.6.1 Postweld heat treatment of nonferrous materials is normally not necessary nor is it desirable. Except as required in paragraphs 6.4.6.2, 6.4.6.3, and 6.4.6.4, postweld heat treatment shall not be performed except by agreement between the purchaser and the Manufacturer. The temperature, time and method of heat treatment shall be covered by agreement.

6.4.6.2 If welded, castings of SB-148, Alloy CDA 954 shall be heat treated after all welding at 620°C-640°C (1150°F-1200°F) for 1.5 hours for the 25 mm (1 in.) of cross section thickness plus 0.5 hour for each additional 25 mm (1 in.) of section thickness. The material shall then be air cooled.

6.4.6.3 Within 14 days after welding, all products of zirconium Grade R60705 shall be heat treated at 500°C-610°C (1000°F-1100°F) for a minimum of 1 hour for thicknesses up to 25 mm (1 in.) plus 0.5 hour for each additional 25 mm (1 in.) of thickness. Above 430°C (800°F) cooling shall be done at a rate not greater than 280°C/hr (500°F/hr) divided by the maximum metal thickness of the shell or head plate in inches but in no case more than 280°C/hr (500°F /hr). From 430°C (800°F), the vessel may be cooled in still air.

6.4.6.4 Postweld Heat Treatment of UNS Numbers N08800, N08810 and N08811 Alloys.

a) Pressure boundary welds and welds to pressure boundaries in vessels with design temperatures above 540°C (1000°F) fabricated from UNS No. N08800 (Alloy 800), UNS No. N08810 (Alloy 800H), or UNS No. N08811 (Alloy 800HT) shall be postweld heat treated. The postweld heat treatment shall consist of heating to a minimum temperature of 885°C (1625°F) for 1.5 hours for thicknesses up to 25 mm (1 in.), and for 1.5 hours plus 0.04 hr/mm (1 hr/in.) of thickness for thicknesses in excess of 25 mm (1 in.). Cooling and heating rates shall be by agreement between the purchaser and Manufacturer. As an alternative, solution annealing in accordance with the material specification is acceptable. Postweld heat treatment of tube-to-tubesheet and expansion bellows attachment welds is neither required nor prohibited.

b) Except as permitted in paragraph 6.4.6.4.c, vessels or parts of vessels that have been postweld heat treated in accordance with the requirements of this paragraph shall again be postweld heat treated after welded repairs have been made.

c) Weld repairs to the weld metal and heat affected zone in welds joining these materials may be made after the final PWHT, but prior to the final hydrostatic test, without additional PWHT. The weld repairs shall meet the requirements shown below.

   1) The Manufacturer shall give prior notification of the repair to the user or to his designated agent and shall not proceed until acceptance has been obtained.

   2) The total repair depth shall not exceed 13 mm (1/2 in.) or 30% of the material thickness, whichever is less. The total depth of a weld repair shall be taken as the sum of the depths for repairs made from both sides of a weld at a given location.

   3) After removal of the defect, the groove shall be examined. The weld repair area must also be examined using the liquid penetrant method in accordance with Part 7.

   4) The vessel shall be hydrostatically tested after making the welded repair.

6.5 Special Requirements For Clad or Weld Overlay Linings, and Lined Parts

6.5.1 Materials

6.5.1.1 Integral or Weld Metal Overlay Clad Base Material or Parts

Integral or weld metal overlay clad base material or parts having applied corrosion resistant linings shall conform to the requirements of paragraph 3.3.6. For the purposes of this Paragraph, the term corrosion resistant includes, but is not limited to, cladding, weld overlay, hard facing, etc., where welding is used to deposit the material.

6.5.1.2 Inserted Strips in Clad Materials

The nominal thickness of inserted strips used to restore cladding at joints shall be equal to the nominal thickness of cladding specified for the plates, backed if necessary with corrosion resistant weld metal deposited in the groove to bring the insert flush with the surface of the adjacent cladding. When insert strips are used, the cladding shall not be considered part of the shell for strength purposes.

6.5.1.3 Weld Metal Composition

Welds that are exposed to the corrosive action of the contents of the vessel should have a resistance to corrosion that is not substantially less than that of the corrosion resistant integral or weld metal overlay cladding or lining. The use of filler metal that will deposit weld metal with practically the same composition as the material joined is recommended. By agreement between the user and Manufacturer, a weld metal of different composition may be used provided it has better mechanical properties and its resistance to corrosion is satisfactory for the intended service.

6.5.2 Joints In Corrosion Resistant Clad or Weld Metal Overlay Linings

The types of joints and welding procedures used shall be selected to minimize the formation of brittle weld composition that may result from the mixture of the corrosion resistant alloy and base material.

NOTE: Because of the different thermal coefficients of expansion of dissimilar metals, caution should be exercised in design and construction under provisions of these paragraphs in order to avoid difficulties in service under extreme temperature conditions or with unusual restraint of parts such as may occur at points of stress concentration.

6.5.3 Welding Procedures

Welding procedures for corrosion resistant clad and weld overlay linings shall be prepared and qualified in accordance with the requirements of Section IX.

6.5.4 Methods to Be Used In Attaching Applied Linings

Applied linings may be attached to the base material and other parts by any method and process of welding that is not excluded by the rules of this Division.

6.5.5 Postweld Heat Treatment of Clad and Lined Weldments

6.5.5.1 Requirements When Base Metal Must Be Postweld Heat Treated

a) Vessels or parts of vessels constructed of corrosion resistant integral or weld metal overlay clad, or applied corrosion resistant lining material shall be postweld heat treated when the base material is required to be postweld heat treated. In applying this requirement, the thickness used to determine the need for postweld heat treatment shall be the total thickness of the base material.

b) When the thickness of the base material requires postweld heat treatment, it shall be performed after the application of corrosion resistant weld metal overlay cladding or applied corrosion resistant lining unless exempted by paragraph 6.4.2.2.

6.5.5.2 Requirements When Base Metal or Lining Is Chromium Alloy Steel

Vessels or parts of vessels constructed of chromium alloy stainless steel clad base material and those lined with chromium alloy stainless steel applied linings shall be postweld heat treated in all thicknesses, except that vessels clad or lined with Type 405 or Type 410S and welded with an austenitic electrode or non-air-hardening nickel-chromium-iron electrode need not be postweld heat treated unless required by paragraph 6.5.5.1.

6.5.6 Requirements for Base Material With Corrosion Resistant Integral or Weld Metal Overlay Cladding

6.5.6.1 Procedure Qualification for Groove Welds in Base Material with Corrosion Resistant Integral or Weld Metal Overlay Cladding

The requirements in Section IX, for procedure qualification shall be followed.

6.5.6.2 When the Integral or Weld Metal Overlay Cladding Thickness is Included in Design Thickness

The welding procedure for groove welds in integral or weld overlay shall be qualified as provided in paragraph

6.2.2.4 when any part of the cladding thickness of the clad base material is included in the design calculations in accordance with paragraph 4.1.9.

6.5.6.3 When the Integral or Weld Metal Overlay Cladding Thickness Is Not Included in Design Thickness

When the cladding thickness is not included in the design calculations (see paragraph 6.5.6.2), the procedure for groove welds shall be qualified as in paragraph 6.5.3. Alternatively, the weld in the base joint or cladding joint may be qualified by itself with the rules in Section IX.

6.5.6.4 Performance Qualification for Groove Welds in Base Material with Corrosion Resistant Integral or Weld Metal Overlay Cladding

Welders and welding operators shall be qualified in accordance with the requirements of Section IX.

6.5.7 Examination Requirements

Examination requirements for vessels with clad or weld metal overlay linings are located in paragraphs 7.4.8.1 and 7.4.8.2.

6.5.8 Inspection and Tests

Inspection and testing requirements for vessels with clad or weld metal overlay linings are located in paragraph 7.4.8.3 and Part 8, respectively.

6.5.9 Stamping and Reports

The provisions for stamping and reports in Part 2 shall apply to vessels that have clad or weld overlay corrosion resistant linings. The Manufacturer's Data Reports shall include the specification and type of lining material and the applicable paragraph under which the shell and heads were designed.

6.6 Special Requirements for Tensile Property Enhanced Q and T Ferritic Steels

6.6.1 General

The following supplemental rules are applicable to steels suitable for welded vessel parts where the tensile properties that have been enhanced by the quenching and tempering heat treatments shown in Table 6.17 and Table 3.A.4. These supplemental rules shall be used in conjunction with the general requirements for fabrication in Part 6, as applicable. The provisions of paragraph 3.10.5.2.d shall also apply to materials whose tensile properties are enhanced by quenching and tempering heat treatment.

6.6.2 Marking on Plates and Other Materials

Any steel stamping shall be done with "low stress" stamps, as commercially available. Steel stamping of all types may be omitted on material below 13 mm (1/2 in.) in thickness. Requirements for the use of other markings in lieu of stamping are covered in paragraph 6.1.1.2.b.

6.6.3 Requirements for Heat Treating After Forming

a) Pieces that are formed after quenching and tempering at a temperature lower than the final tempering temperature shall be heat treated in accordance with Table 6.17 when the extreme fiber elongation from forming exceeds 5% as determined by the lesser of the applicable equations in Table 6.1.

b) Pieces formed at temperatures equal to or higher than the original tempering temperature shall be re-quenched and tempered in accordance with the applicable material specifications either before or after welding into the vessel.

6.6.4 Minimum Thickness after Forming

The minimum thickness after forming of any section subject to pressure shall be 1.6 mm (1/16 in.).

6.6.5 Welding Requirements

6.6.5.1 Qualification of Welding Procedures and Welders

The qualification of the welding procedure and the welders shall conform to the requirements of Section IX and such qualification tests shall be performed on postweld heat treated specimens when a postweld heat treatment is used.

6.6.5.2 Additional Welding Requirements

a) Filler metal containing more than 0.06% vanadium shall not be used for weldments subject to postweld heat treatment.

b) The materials in Table 6.18 are exempt from production impact tests of the weld metal in accordance with paragraph 3.11 under the conditions given below:

   1) One of the high nickel alloy filler metals in Table 6.19 is used.

   2) All required impact tests shall be performed as part of the procedure qualification tests as specified in paragraph 3.11.

   3) Production impact tests of the heat affected zone shall be performed in accordance with paragraph 3.11.

   4) The welding processes are limited to gas metal arc, shielded metal arc, and gas tungsten arc.

   5) The minimum design metal temperature of the vessel shall be not colder than -195°C (-320°F).

c) For SA-508 and SA-543 materials, the following, in addition to the variables in Section IX, QW-250, shall be considered as essential variables requiring requalification of the welding procedure.

   1) A change in filler metal SFA classification or to weld metal not covered by an SFA specification.

   2) An increase in the maximum interpass temperature or a decrease in the minimum specified preheat temperature. The specified range between the preheat temperature and the interpass temperature shall not exceed 85°C (150°F).

   3) A change in the heat treatment (procedure qualification tests shall be subjected to heat treatment essentially equivalent to that encountered in fabrication of the vessel or vessel parts, including the maximum total aggregate time at temperature or temperatures and cooling rates).

   4) A change in the type of current (AC or DC), polarity, or a change in the specified range for amp, volt, or travel speed.

   5) A change in the thickness, t, of the welding procedure qualification test plate as follows:

      i) For welded joints that are quenched and tempered after welding, any increase in thickness. The minimum thickness qualified in all cases is 6 mm (1/4 in.).

      ii) For welded joints with a thickness, t, less than 16 mm (5/8 in.) that are not quenched and tempered after welding, for any decrease in thickness. The maximum thickness qualified is $2t$.

      iii) For welded joints with a thickness, t, greater than or equal to 16 mm (5/8 in.) that are not quenched and tempered after welding, for any departure from the range of 16 mm (5/8 in.) to $2t$.

6) Consumables control, drying, storage, and exposure requirements shall be in accordance with the following.

   i) Due consideration shall be given to protection of electrodes and fluxes for all welding processes in order to minimize moisture absorption and surface contamination.

   ii) Electrodes for shielded metal arc welding shall be low hydrogen type conforming to SFA-5.5 Electrodes shall be purchased or conditioned so as to have a coating moisture content not greater than 0.2% by weight. Once opened, electrode storage and handling shall be controlled so as to minimize absorption of moisture from the ambient atmosphere. The practice used for controlling the moisture content shall be developed by the vessel Manufacturer or that recommended by the electrode manufacturer.

7) The minimum preheat temperature is a function of the material thickness, t, as shown below. The preheat temperature shall be maintained for a minimum of 2 hours after completion of the weld joint.

   i) 40°C (100°F) for $t \le 13$ mm (1/2 in.)

   ii) 95°C (200°F) for 13 mm (½ in.) < $t \le 38$ mm (1-1/2 in.)

   iii) 150°C (300°F) for $t>38$ mm (1-1/2 in.)

   iv) For SA-517and SA-592 materials, the requirements of paragraphs 6.6.5.2.c.1 through 6.6.5.2.c.4 and 6.6.5.2.c.6 in addition to the variables in Section IX, QW-250, shall be considered as essential variables requiring requalification of the welding procedure.

8) The postweld heat treatment temperature as required by Table 6.17 may be waived for SA-517 and SA-592 materials with a nominal thickness, t, such that, 15 mm (9/16) < $t \le 32$ mm (1-1/4 in.) provided all of the following conditions are met:

   i) A minimum preheat of 95°C (200°F) and a maximum interpass of 205°C (400°F) are used.

   ii) After completion of welding and without allowing the weldment to cool below the minimum preheat temperature, the temperature of the weldment is raised to a minimum of 205°C (400°F) and maintained at that temperature for at least 4 hr; and

   iii) All welds are examined by nondestructive examination in accordance with the provisions of this Part.

6.6.5.3 Preparation of Base Metal

The preparation of plate edges, welding bevels, and chamfers and similar operations involving the removal of metal shall be by machining, chipping, or grinding, by gas cutting or gouging. If metal removal is accomplished by methods involving melting, such as gas cutting or arc air gouging, the metal removal shall be done with appropriate precautions to avoid cracking. When thermal cutting is used, the effect on mechanical properties shall be taken into consideration. The edges to be welded shall be uniform and smooth.

6.6.5.4 Joint Alignment

a) Longitudinal Joint Alignment – In lieu of paragraph 6.1.6.1, the longitudinal joint misalignment for quenched and tempered steels shall not exceed 20% of the nominal plate thickness or 2.5 mm (3/32 in.).

b) Circumferential Joint Alignment – In lieu of paragraph 6.1.6.1, the circumferential joint misalignment for quenched and tempered steels shall not exceed the following values.

   1) If $t \le 24$ mm (15/16 in.), then the tolerance is 20% of the plate thickness.

   2) If 24 mm (15/16 in.) < $t \le 38$ mm (1-1/2 in.), then the tolerance is 5 mm (3/16 in.).

   3) If $t > 38$ mm (1-1/2 in.), then the tolerance is 12.5% of the nominal plate thickness but not more than 6 mm (1/4 in.)

6.6.5.5 Weld Finish

The requirements of paragraphs 6.2.4.1 and 7.5.2 shall be met, except that the maximum weld reinforcement shall not exceed 10% of the plate thickness or 3 mm (1/8 in.), whichever is less. The edge of the weld deposits shall merge smoothly into the base metal without undercuts or abrupt transitions. This requirement shall apply to fillet and groove welds as well as to butt welds.

6.6.5.6 Attachment and Temporary Welds

a) Material for Structural Attachments and Stiffening Rings – Except as permitted by paragraph 4.2.5.6, all permanent structural attachments and stiffening rings that are welded directly to pressure parts shall be made of material whose specified minimum yield strength is within ±20% of that of the material to which they are attached.
b) Fabrication of Structural and Temporary Welds – Welds for pads, lifting lugs, and other nonpressure parts, as well as temporary lugs for alignment, shall be made by qualified welders in full compliance with a qualified welding procedure. The type of welds used shall conform to the requirements of paragraph 4.2.5.6. Examination requirements for these welds are covered in paragraph 6.6.8.
c) Removal of Temporary Welds – Temporary welds shall be removed and the metal surface shall be restored to a smooth contour. The area shall be examined by a magnetic particle or liquid penetrant method for the detection and elimination of cracks or crack-like defects. If repair welding is required, it shall be in accordance with a qualified procedure, and the finished weld surface shall be examined in accordance with Part 7. Temporary welds and repair welds shall be considered the same as all other welds insofar as requirements for qualified welders and welding procedures and for heat treatment are concerned.

6.6.6 Postweld Heat Treatment

6.6.6.1 Vessels or parts of vessels constructed of quenched and tempered steels shall be postweld heat treated when required in Table 6.17, when welding ferritic materials greater than 3 mm (1/8 in.) thick with the electron beam welding process, and when welding materials of all thicknesses using the inertia and continuous drive friction welding processes. The total thickness of the base material shall be used as the thickness used to determine postweld heat treatment requirements for clad or weld overlayed parts.

6.6.6.2 Requirements for Postweld Heat Treatment

Postweld heat treatment shall be performed in accordance with paragraph 6.4, as modified by the requirements of Table 6.17 and paragraph 6.6.6. In no case shall the PWHT temperature exceed the tempering temperature. PWHT and tempering may be accomplished concurrently. The maximum cooling rate established in paragraph 6.4.4.e need not apply. Where accelerated cooling from the tempering temperature is required by the material specification, the same minimum cooling rate shall apply to PWHT.

6.6.6.3 Postweld Heat Treatment of Connections and Attachments

All welding of connections and attachments shall be postweld heat treated whenever required by Table 6.17 based on the greatest thickness of material at the point of attachment to the head or shell (see paragraphs 6.4.2.5 and 6.4.2.6).

6.6.6.4 Heat Treatment Procedure

a) Heating Furnace – Furnaces for heating and for quenching and tempering shall be provided with suitable equipment for the automatic recording of temperatures. The metal temperature of the vessel or vessel part during the holding period shall be recorded and shall be controlled within ±15°C (±25°F).
b) Liquid Quenching of Flat Plates or Parts – Liquid quenching of flat plates and individual parts shall be done as required by the applicable material specifications.
c) Quenching of Shell Sections or Heads – Formed plates for shell sections and heads may be quenched by sprays or immersion.
d) Quenching of Entire Vessels – Entire vessels, after completion of all welding operations, may be quenched by sprays or immersion.

6.6.6.5 Design and Operation of Quenching Equipment

The design and operation of spray equipment and the size of tanks and provision for forced circulation shall be selected to produce a severity of quench in the quenched item sufficient to meet, in representative test specimens after tempering, the requirements of the material specifications.

6.6.7 Heat Treatment Certification Tests

6.6.7.1 Heat Treatment Verification Tests

Tests shall be made to verify that the quenching and tempering heat treatments and subsequent thermal treatments, performed by the Manufacturer, have produced the required properties. The requirements of paragraph 6.6.7.2.b and 6.6.7.2.c shall be taken as minimum steps toward these objectives.

6.6.7.2 Certification Test Procedure

a) One or more test coupons representative of the material and the welding in each vessel or vessel component shall be heat treated with the vessel or vessel component.

b) One or more test coupons from each lot of material in each vessel (see paragraph 6.6.7.2.e) shall be quenched with the vessel or vessel component. A lot is defined as material from the same heat, quenched or normalized simultaneously and whose thicknesses are within ±20% or 13 mm (1/2 in.) of nominal thickness, whichever is smaller. The test coupons shall be so proportioned that tensile and impact tests may be taken from the same locations relative to thickness as are required by the applicable material specifications. Weld metal and heat affected zone impact test specimens shall be taken from locations relative to coupon thickness in accordance with paragraph 3.11.8.2. The gage length of tensile specimens and the middle third of the length of impact specimens must be located at a minimum distance of $1xt$ from the quenched edge and/or end of the test coupon, where t is the thickness of the material that the test coupon represents. If desired, the effect of this distance may be achieved by temporary attachment of suitable thermal buffers. The effectiveness of such buffers shall be demonstrated by tests.

c) In cases where the test coupon is not attached to the part being treated, it shall be quenched from the same heat treatment charge and under the same conditions as the part which it represents (see paragraph 3.10.5). It shall be so proportioned that test specimens may be taken from the locations prescribed in paragraph 6.6.7.2.b.

d) Tempering Requirements

   1) Attached Test Coupons – The test coupons shall remain attached to the vessel or vessel component during tempering, except that any thermal buffers may be removed after quenching. After the tempering operation and after removal from the component, the coupon shall be subjected to the same thermal treatment(s), if any, to which the vessel or vessel component will be later subjected. The holding time at temperature shall not be less than that applied to the vessel or vessel component (except that the total time at each temperature may be applied in one heating cycle) and the cooling rate shall be no faster.

   2) Separate Test Coupons – Test coupons that are quenched separately, as described in paragraph 6.6.7.2.c, shall be tempered similarly and simultaneously with the vessel or component that they represent. The conditions for subjecting the test coupons to subsequent thermal treatment(s) shall be as described in paragraph 6.6.7.2.b.

e) Number of Tests – For base materials, one tensile and one impact test shall be made on material from coupons representing each heat-treated lot of material in each vessel or vessel component. A lot is defined as material from the same heat quenched simultaneously and whose thicknesses are within ±20% or 13 mm (1/2 in.) of nominal thickness, whichever is smaller.

   1) Coupons not containing welds shall meet the complete tensile requirements of the material specification and impact requirements of this Part.

   2) Coupons containing weld metal shall be tested across the weld and shall meet the ultimate tensile strength requirements of the material specifications; in addition, the minimum impact requirements shall be met by samples with notches in the weld metal. The form and dimension of the tensile test

specimen shall conform to QW-462.1(a) or QW-462.1(d) of Section IX. The yield strength and elongation are not a requirement of this test. Charpy impact testing shall be in accordance with the requirements of paragraph 3.11.

6.6.8 Examination Requirements

Examination requirements for vessels and vessel parts constructed of quenched and tempered steels are located in paragraphs 7.4.9.

6.6.9 Inspection and Tests

The provisions for inspection and testing in Part 7 and Part 8, respectively, shall apply to vessels and vessel parts constructed of quenched and tempered steels.

6.6.10 Stamping and Reports

The provisions for stamping and reports in Part 2 shall apply to pressure vessels constructed in whole or in part of quenched and tempered steels, except that the use of nameplates is mandatory for shell thicknesses below 13 mm (1/2 in.). Nameplates are preferred on vessels of quenched and tempered steels in thicknesses above 13 mm (1/2 in.) instead of stamping. In addition to the required marking, the letters QT shall be applied below the symbol.

6.7 Special Requirements for Forged Fabrication

6.7.1 General

The rules in the following paragraphs apply specifically to vessels, main sections of vessels and other vessel parts, and to liquid quenched and tempered, integrally forged vessels without welded joints, and shall be used to supplement the applicable requirements for fabrication given in paragraph 6.1. For high alloy steel forged vessels, the applicable paragraphs of Part 3 shall also apply.

6.7.2 Ultrasonic Examination

Ultrasonic examination requirements for vessels and vessel parts constructed from forged fabrication are located in paragraph 7.4.10.1.

6.7.3 Toughness Requirements

a) For vessels constructed of SA-372 Grade J, Class 110 material, transverse impact tests shall be made at the minimum allowable temperature in accordance with paragraph 3.11, except that in no case shall the test temperature be warmer than -30°C (-20°F). Paragraph 3.11.2.1.b and paragraph 3.11.3.3 are not applicable. Certification is required.
b) For vessels constructed of SA-723 Class 1, Grades 1, 2, and 3, or SA-723 Class 2, Grades 1, 2, and 3 materials, the impact requirements of paragraph 3.11.3 shall be met when tested at 4°C (40°F) maximum.

6.7.4 Tolerances on Cylindrical Forgings

6.7.4.1 Localized Thin Areas

Forgings are permitted to have small areas thinner than required if the adjacent areas surrounding each has sufficient thickness to satisfy the provisions of Part 5.

6.7.4.2 Tolerances on Body Forgings

a) Correction of Surface Irregularities to Meet Tolerances

   Irregularities in the surface under consideration may be corrected by welding or other means to meet these tolerances. If welding is done, then it shall meet the requirements of paragraph 6.7.7.
b) Use of Out-of-Round Forgings for Lower Pressure

   If out-of-roundness exceeds the limit in paragraph 6.1.2.7 and the condition cannot be corrected, then

the forging shall be rejected, except that if the out-of-roundness does not exceed 3%, the forging may be certified for a reduced pressure, P^*, calculated using Equations (6.1) and (6.2). The measurements used in this equation shall be corrected for the specified corrosion allowance.

$$P^* = P\left(\frac{1.25}{S_b/S \;+1}\right) \qquad when\ S_b \geq 0.25S \tag{6.1}$$

$$P^* = P \qquad when\ S_b < 0.25S \tag{6.2}$$

$$S_b = \frac{1.5PR_1t\left(D_1 - D_2\right)}{t^3 + 3\left(\dfrac{P}{E_y}\right)\left(R_1R_a^2\right)} \tag{6.3}$$

$$R_1 = \frac{D_1 + D_2}{4} \tag{6.4}$$

$$R_a = R_1 + \frac{t}{2} \tag{6.5}$$

6.7.5 Methods of Forming Forged Heads

6.7.5.1 Heads shall be made as separate forgings or by closing the extremities of a hollow forged body to such shape and dimensions as may be required to produce the final form desired.

6.7.5.2 Tolerances on Head Forgings

Tolerances shall meet the requirements of paragraph 6.1.2.7.

6.7.5.3 Correction of Surface Irregularities to Meet Tolerances

Irregularities may be corrected in accordance with paragraph 6.7.4.2.a.

6.7.6 Heat Treatment Requirements for Forged Fabrication

6.7.6.1 Heat Treatment When Vessels Are Fabricated by Welding

Vessels fabricated by welding of forged parts requiring heat treatment shall be heat treated in accordance with the applicable material specification:

a) after all welding is completed, or
b) prior to welding, followed by postweld heat treatment of the finished weld in accordance with paragraph 6.4.2.

6.7.6.2 Heat Treatment When Material Is to Be Normalized or Annealed

After all forging is completed, each vessel or forged part fabricated without welding shall be heat treated in accordance with the applicable material specification. When irregularities are corrected by welding, subsequent heat treatment shall be in accordance with paragraph 6.7.8.3.b.

6.7.6.3 Heat Treatment of Quenched and Tempered Ferritic Material

Vessels fabricated of SA-372 forging material to be liquid quenched and tempered shall be subjected to this heat treatment in accordance with the applicable material specifications after all forging and welding is completed, except for seal welding of threaded openings, which may be performed either before or after final heat treatment.

a) Examination of Quenched and Tempered Vessels – After the final heat treatment, quenched and tempered vessels shall be examined for the presence of cracks on the outside surface of the shell and heads and on the inside surface where practicable. This examination shall be made by the liquid penetrant or a magnetic particle method in accordance with Part 7.

b) Check of Heat Treatment by Hardness Testing – After the final heat treatment, liquid quenched and tempered forgings, except those made of austenitic steels, shall be subjected to Brinnell hardness tests at 1.5 m (5 ft.) intervals with a minimum of four readings at each of not less than three different locations representing approximately the center and each end of the heat treated forgings. The average of the individual Brinnell hardness numbers at each location shall be not more than 10% below or more than 25% above the number corresponding to the specified minimum tensile strength of the material. The highest average hardness number shall not exceed the lowest average value on an individual vessel by more than 40. Other hardness testing methods may be used and converted to Brinnell numbers by means of the table in ASTM E 140.

6.7.6.4 Heat Treatment of Austenitic Material

In the case of austenitic steels, the heat treatment procedures followed shall be in accordance with paragraph 6.4.

6.7.6.5 Ferrous Material Not Requiring PWHT

Postweld heat treatment of vessels fabricated by welding of forged parts not requiring heat treatment shall meet the requirements of paragraph 6.4.

6.7.7 Welding For Fabrication

6.7.7.1 All welding used in connection with the fabrication of forged vessels or components shall comply with the applicable requirements of paragraph 6.2 except as modified in paragraph 6.7.7.2.

6.7.7.2 Restrictions on Ferrous Materials with Carbon Content Exceeding 0.35%

When the carbon content of the material exceeds 0.35% by heat analysis, the vessel shall be fabricated without welding, except for repairs in accordance with paragraph 6.7.8.2, for minor nonpressure attachments, for seal welding of threaded openings limited to fillet welds of not over 6 mm (1/4 in.) throat dimension, and for adding reinforcement to threaded, flanged, or studded openings. Such welding shall be allowed under the following conditions.

a) The suitability of the electrode and procedure, including preheat and post-heat, shall be established by making a groove weld specimen as shown in QW-461.2 and QW-461.3 of Section IX in material of the same analysis and of thickness in conformance with QW-451.1 and QW-451.2 of Section IX. The specimen before welding shall be in the same condition of heat treatment as the work it represents, and after welding the specimen shall be subjected to heat treatment equivalent to that contemplated for the work. Tension and bend specimens, as shown in QW-462.1, QW-462.2, and QW-462.3a of Section IX, shall be made. These tests shall meet the requirements of QW-150 and QW-160 of Section IX. The radius of the mandrel used in the guided bend test shall be as shown in Table 6.20.

b) Welders shall be qualified for minor nonpressure attachments, seal welds, and for fillet welds specified by making and testing a specimen in accordance with QW-462.4(b) and QW-180 of Section IX. Welders shall be qualified for adding weld reinforcement for openings and for repair welding by making a test plate in accordance with QW-461, from which the bend tests outlined in QW-452.1 and QW-452.2 of Section IX shall be made. The electrode used in making these tests shall be of the same AWS classification as that specified in the procedure. The material for these tests can be carbon steel plate or pipe, provided the test specimens are preheated, welded, and postweld heat treated in accordance with the procedure specification for the type of electrode involved.

c) The finished welds shall be postweld heat treated or given a further heat treatment as required by the applicable material specification. The welding permitted in paragraph 6.7.7.2 shall be performed prior to final heat treatment except for seal welding of threaded openings, which may be performed either before or after final heat treatment.

d) The finished welds shall be examined after postweld heat treatment by the liquid penetrant method or magnetic particle method in accordance with Part 7.

6.7.8 Repair of Defects in Material

6.7.8.1 Removal of Surface Defects

Surface imperfections such as chip marks, blemishes, or other irregularities may be removed by grinding or machining, and the surface exposed shall be blended smoothly into the adjacent surface when sufficient wall thickness permits thin areas in compliance with the requirements of paragraph 6.7.4.1.

6.7.8.2 Repair of Defects by Welding

Thinning to remove defects may be repaired by welding only after approval by the Inspector. Defects shall be removed to sound metal and verified using an acid etch or any other suitable method of examination. The welding shall meet the requirements of paragraphs 6.7.8.3 and 6.7.8.4.

6.7.8.3 Weld Repairs of Material Containing 0.35% Carbon or Less

Material having carbon content of 0.35% or less (by heat analysis) may be repaired by welding provided the requirements of paragraph 6.7 are met.

a) Qualification of Welding Procedure and Welders – The welding procedure and welders shall be qualified in accordance with Section IX.

b) Postweld Heat Treatment – Postweld heat treatment after welding shall be governed as follows.

   1) All welding shall be postweld heat treated if required by paragraph 6.4.2.

   2) Fillet welds need not be postweld heat treated unless required by paragraph 6.7.8.3.b.1 or unless the fillet welds exceed the limits given in paragraph 6.4.2, in which case they shall be heat treated in accordance with the requirements of this paragraph.

   3) Repair welding shall be postweld heat treated when required by paragraph 6.7.8.3.b.1, if it exceeds 3780 mm^2 (6 $in.^2$) at any spot, or if the maximum depth exceeds 6 mm (1/4 in.).

c) Examination of Weld Repairs – See paragraph 7.4.10.2.a.

6.7.8.4 Weld Repairs of Material Containing More Than 0.35% Carbon

Material having carbon content over 0.35% (by heat analysis) may be repaired by welding when the requirements of this paragraph are met.

a) Qualification of Welding Procedure and Welders – The welding procedure and welders shall be qualified in accordance with Section IX and the additional requirements of paragraph 6.7.7.2.a and 6.7.7.2.b.

b) Postweld Heat Treatment – The finished repair welds shall be postweld heat treated or given a further heat treatment as required by the applicable material specification.

c) Examination of Weld Repairs – see paragraph 7.4.10.2.b.

6.7.8.5 Repair of Weld Defects

a) The repair of welds in forgings having carbon content not exceeding 0.35% by heat analysis shall follow the requirements of paragraph 6.2.7.

b) The repair of welds in forgings having a carbon content exceeding 0.35% by heat analysis shall follow the requirements of 6.7.8.4.

6.7.9 Threaded Connections to Vessel Walls, Forged Necks, and Heads

6.7.9.1 Requirements for Straight Threaded Openings

Straight threaded openings shall meet the rules governing openings and reinforcements in paragraph 4.5, except as limited in paragraph 6.7.9.2. The length of thread shall be calculated for the opening design.

6.7.9.2 Location and Maximum Size of Straight Threaded Openings

Straight threaded center openings in integrally forged heads with nozzle extensions shall not exceed the smaller of one-half the vessel diameter or DN 200 (NPS 8) as shown in Figure 6.3.

6.7.9.3 Requirements for Tapered Threaded Openings

Tapered threaded openings shall meet the limitations and requirements of paragraph 4.5.

6.7.9.4 Seal Welding of Threaded Openings

When piping or fittings are installed in threaded openings and seal welding is employed, the work shall be performed and examined at the vessel Manufacturer's plant and included in the certification. Seal welding shall comply with paragraph 6.7.7.

6.7.10 Inspection, Examination, and Testing

6.7.10.1 The rules in the following paragraphs apply specifically to the inspection, examination, and testing of forged vessels and their component parts. These rules shall be used to supplement the applicable requirements and examination in Part 7.

a) All forged vessels shall be examined as manufacturing proceeds to ensure freedom from loose scale, gouges or grooves, and cracks or seams. After fabrication has passed the machining stage, the vessel body shall be measured at suitable intervals along its length to get a record of variations in wall thickness, and the nozzles for connecting piping and other important details shall be checked for conformity to the design dimensions.

b) Surfaces that are not to be machined shall be carefully examined for visible defects such as seams, laps, or folds. On surfaces to be machined, the examination shall be made after machining. Regions from which defective material has been removed shall be examined after removal and again after any necessary repair.

6.7.10.2 Forged Parts

a) Partial Data Reports Required – When welding is used in the fabrication of forged parts completed elsewhere, the manufacturer of the forged parts shall furnish a Partial Data Report, Form A-2.

b) Identification and Certification – All parts forgings completed elsewhere shall be marked with the forging manufacturer's name and the forging identification, including material designation. Should identifying marks be obliterated in the fabrication process, and for small parts, other means of identification shall be used. The forging manufacturer shall furnish reports of chemical and mechanical properties of the material and certification that each forging conforms to all requirements of Part 3.

c) Welded Repairs and Their Certification – Welded repairs to parts forgings need not be inspected by an Authorized Inspector at the plant of the forging manufacturer, but the forging manufacturer shall obtain the approval of the vessel Manufacturer and furnish a report of the extent and location of such repairs, together with certification that they were made in accordance with all other requirements of paragraph 6.7.8.4, as applicable. If desired, welding repairs of forgings made elsewhere may be made, examined,

and tested at the shop of the vessel Manufacturer.

6.7.10.3 Check of Heat Treatment and Postweld Heat Treatment

The Inspector shall check the provisions made for heat treatment to ensure that the heat treatment is carried out in accordance with the provisions of paragraph 6.7.6. The Inspector shall also ensure that postweld heat treatment is done after repair welding when required under the rules of paragraph 6.7.8.3.b.

6.7.10.4 Inspection of Test Specimens and Witnessing Tests

a) Test Specimens – When test specimens are to be taken under the applicable material specifications, the Inspector may witness the selection, identifying stamping, and testing of these specimens.

b) Tests and Retests – Tests and retests shall be made in accordance with the requirements of the material specification.

6.7.11 Stamping and Reports for Forged Vessels

6.7.11.1 Stamping Requirements

The rules of Part 2 shall apply to forged vessels as far as practicable. Vessels constructed of liquid quenched and tempered material, other than austenitic steels, shall be stamped on the thickened head, using low stress stamps as commercially available unless a nameplate is used.

6.7.11.2 Information Required on Data Reports for Integrally Forged Vessels

Data reports for integrally forged vessels shall include the heat number or numbers of the metal in the ingot from which the vessel was forged and the test results obtained for the forging.

6.7.12 Pressure Relief Devices

The provisions for pressure relief devices of Part 9 shall apply without supplement.

6.8 Special Fabrication Requirements for Layered Vessels

6.8.1 General

The rules in the following paragraphs apply specifically to layered shells, layered heads, and layered transition sections and shall be used to supplement, or be used in lieu of, the applicable requirements given in paragraphs 6.1 through 6.6. Where requirements differ from those of paragraphs 6.2, 6.4, and 6.6, they are specifically delineated.

6.8.2 General Fabrication Requirements

Requirements shall be in accordance with paragraph 6.1. For layered vessels, the minimum thickness permitted for layers is 3 mm (1/8 in.).

6.8.3 Welding Fabrication Requirements

The welding fabrication shall be in accordance with paragraph 6.2, except that the welding procedure qualification requirements are modified for layered construction as given in paragraph 6.8. Also the specified requirements are modified for welded joints in paragraph 4.13 and for nondestructive examination in paragraph 6.8.6.

6.8.4 Welding Qualification and Records

6.8.4.1 Requirements for welding qualification and records shall be in accordance with paragraph 6.2.2, except that the layered test plate welding procedure qualification of Section IX in paragraphs 6.2.2.1 and 6.2.2.4 shall be as modified in paragraph 6.8.4.2.

6.8.4.2 Welding Procedure Qualification

a) The minimum and maximum thicknesses qualified by procedure qualification test plates shall be as shown in Table QW-451 of Section IX, except that:

1) For the longitudinal joints of the layer section of the shell, the qualification shall be based upon the thickness of the thickest individual layer exclusive of the inner shell or inner head.

2) For circumferential joint procedure qualification, the thickness of the layered test plate need not exceed 75 mm (3 in.), shall consist of at least 2 layers, but shall not be less than 50 mm (2 in.) in thickness.

3) For circumferential weld joints made individually for single layers and spaced at least one layer thickness apart, the procedure qualification for the longitudinal joint applies.

b) The longitudinal weld joint of the inner shell or inner head and the longitudinal weld joint of the layer shell or layer head shall be qualified separately except if of the same P-Number material. The weld gap of the longitudinal layer weld joint shall be the minimum width used in the procedure qualification for layers 22 mm (7/8 in.) and less in thickness.

c) The circumferential weld joint of the layer to layer shell or to layer head shall be qualified with a simulated layer test plate as shown in Figure 6.4 for layer thicknesses 22 mm (7/8 in.) and under. A special type of joint tensile specimen shall be made from the layer test coupon as shown in Figure 6.5. Face and root bend specimens shall be made of both the inner and outer weld to the thickness of the layer by cutting the weld to the layer thickness.

d) The circumferential weld joint of the layer shell for layer thicknesses 22 mm (7/8 in.) and under to the solid head, flange, or end closure shall be qualified with a simulated layer test coupon as shown in Figure 6.4, wherein one side of the test coupon is solid throughout its entire thickness. A special type of joint tensile specimen shall be made from the test coupon as shown in Figure 6.5. Face and root bend specimens shall be made of both the inner and outer weld to the thickness of the layer by slicing the weld and solid portion to the layer thickness.

6.8.4.3 Welding Performance Qualification

Welding shall be performed only by welders and welding operators who have been qualified in accordance with Section IX. The minimum and maximum thicknesses qualified by any welder test plate shall be as shown in Table QW-452 of Section IX.

6.8.5 Specific Requirements for Welded Joints

6.8.5.1 The rules of the following paragraphs shall be used in lieu of paragraphs 6.2.4 and 6.2.6.

6.8.5.2 Welding of Joints

Paragraph 4.13.6 covers the types of joints permitted, according to location, in layered vessels and their components. Paragraph 4.13.7 covers rules for attaching nozzles and other pressure connections by welding. Paragraph 4.2.5.6 and paragraph 4.13.10 provide rules for attaching nonpressure parts and stiffeners. Examination requirements are summarized in Table 7.4.

6.8.5.3 Type No.1 Butt Joints

a) Type No. 1 butt joints are defined in paragraph 4.2.3 and Table 4.2.2.

b) Type No. 1 butt joints shall have complete joint penetration and full fusion and shall be free from undercuts, overlaps, or abrupt ridges or valleys (See Table 7.6, No. 6). To assure that the weld grooves are completely filled so that the surface of the weld metal at any point does not fall below the surface of the adjoining plate, weld metal may be built up as reinforcement on both sides of the plate. The thickness of the reinforcement on each side of the plate shall not exceed the limits specified in paragraph 6.2.4.1.d.

6.8.5.4 Type No.2 Butt Joints

a) Type No.2 butt joints are defined in paragraph 4.2.3 and Table 4.2.2.

b) When Type No.2 butt joints are used, the components to be joined shall be aligned and separated so that there will be complete penetration and fusion at the bottom of the joints for their full length. However, for assuring complete filling of the weld grooves, weld reinforcement need be supplied only on the side opposite the backing strip. Weld reinforcement need not be provided on welds which are subsequently ground flush.

c) Backing strips shall be continuous and any splices shall be butt welded. Circumferential single-welded butt joints with one plate offset to form a backing strip are prohibited.

6.8.5.5 Fillet Welded Joints

a) Fillet welded joints are defined in paragraph 4.2.5.1.d and Table 4.2.2.

b) The surface of fillet welds shall be free from coarse ripples or grooves, undercuts, overlaps, and abrupt ridges or valleys, and shall merge smoothly with the surfaces joined.

6.8.5.6 Welds Attaching Nonpressure Parts and Stiffeners

The rules governing the types of welds which may be used to join supports, lugs, brackets, stiffeners, and other attachments to the vessel wall are set forth in paragraph 4.2.5.6.

6.8.5.7 Surface Weld Metal Buildup

a) Construction in which deposits of weld metal are applied to the surface of base metal for the purpose of restoring the thickness of the base metal for strength consideration or modifying the configuration of weld joints in order to provide the tapered transition requirements of paragraph 4.2 or paragraph 6.1.6.2 in solid wall sections shall be performed in accordance with the requirements of paragraphs 6.2.4.9. Details for using layers as transitions are covered in paragraph 4.13.

b) A butt welding procedure qualification in accordance with the provisions of Section IX shall be performed for the thickness of weld metal deposited, prior to production welding.

6.8.6 Nondestructive Examination of Welded Joints

Nondestructive examination requirements for layered vessels are located in paragraph 7.4.11.

6.8.7 Welded Joint Efficiency

If the nondestructive examination outlined in paragraph 6.8.6 is complied with, the weld joint efficiency for design purposes shall be 100%.

6.8.8 Contact between Layers

6.8.8.1 Requirements for contact between layers are covered in paragraph 4.13.12.1.

6.8.8.2 Alternative to Measuring Contact between Layers during Construction

An alternative to measuring the contact between layers during construction is provided in paragraph 4.13.12.2.

6.8.8.3 Rules for Calculating Maximum Permissible Gaps

Rules for computing permissible gaps are provided in paragraph 4.13.12.3.

6.8.9 Vent Holes

Vent holes shall be provided to detect leakage of the inner shell and to prevent buildup of pressure within the layers as follows.

a) In each shell course or head segment, a layer may be made up of one or more plates. Each layer plate shall have at least two vent holes 6 mm (1/4 in.) minimum diameter. The vents holes may be drilled radially through the multiple layers or may be staggered in individual layer plates.

b) For continuous coil wrapped layers, each layered section shall have at least four vent holes 6 mm (1/4 in.) minimum diameter. Two of these vent holes shall be located near each end of the section and spaced approximately 180° apart.

c) The minimum requirement for spirally wound strip layered construction shall be 6 mm (1/4 in.) minimum diameter vent holes drilled near both edges of the strip. These vent holes shall be spaced for the full length of the strip and shall be located a distance of approximately $\pi R_m / \tan\theta$. If a strip weld covers a vent hole, partially or totally, an additional vent hole shall be drilled on each side of the obstructed hole. In addition to the above, holes may be drilled radially through the multiple layers.

d) Vent holes shall not be obstructed. If a monitoring system is used, it shall be designed to prevent buildup of pressure within the layers.

6.8.10 Heat Treatment of Weldments

a) When required, pressure parts shall be postweld heat treated in accordance with paragraph 6.4 and paragraph 6.6; however, the completed layered vessels or layered vessel sections need not be postweld heat treated provided the requirements of paragraph 6.8.10.b are satisfied.

b) Unless required by paragraph 6.4.2, completed layered vessels or layered vessel sections need not be postweld heat treated when welded joints connect a layered section to a layered section, or a layered section to a solid wall, provided all of the following conditions are met.

   1) The thickness referred to in paragraph 6.4.2.7 or paragraph 6.6.6 is the thickness of one layer. Should more than one layer be used, the thickness of the thickest layer shall govern.

   2) The finished joint preparation of a solid section or solid nozzle that is required to be postweld heat treated under the provisions of paragraph 6.4.2.6 or paragraph 6.6.6 shall be provided with a buttered layer (i.e. built-up overlay welding) of at least 3 mm (1/8 in.) thick of weld not requiring postweld heat treatment. Solid sections constructed of P-No. 1 materials need not have this buttered layer. Postweld heat treatment of the buttered solid section shall then be performed prior to attaching to the layered sections. Postweld heat treatment following attachment to the layered section is not required unless the layered section is required to be postweld heat treated.

   3) The multipass welding technique is used and the weld layer thickness is limited to 10 mm (3/8 in.) maximum. When quenched and tempered materials are used (see Table 3.A.4), the last pass shall be completed using a temper bead welding technique, except for 5%, 8%, and 9% nickel steels. The temper bead welding treatment is done when the final beads of welding are made over-flush, deposited only on previous beads of welding for tempering purposes without making contact with the base metal, and then removing these final beads.

c) The postweld heat treating rules in paragraph 6.8.10 shall apply to all weld repairs.

6.9 Nomenclature

d original inside diameter.
d_f final inside diameter.
d_p peaking dimension.
D inside diameter.
D_f final outside diameter.
D_b diameter of the blank plate or the diameter of the intermediate product.
D_o original outside diameter.
D_1 maximum inside diameter.
D_2 minimum inside diameter.
E_y modulus of elasticity at the service temperature, see Annex 3.D.
ε_f calculated forming strain.
ε_L longitudinal contraction.
ε_T tangential contraction.
g depth of blend grind
L initial length.
L_f final length.
P design pressure.
P^* reduced operating pressure due to out-of-roundness.
r nominal outside radius of pipe or tube or blend grind radius.
R_a average radius to middle of the shell wall at critical section.
R_o original mean radius, equal to infinity for a flat plate.
R_f final mean radius.
R_o original mean radius, equal to infinity for a flat plate.
R_1 Average inside radius at critical section.
S Allowable stress from Annex 3.A evaluated at the design temperature.
S_b bending stress at the service temperature due to out-of-roundness.
t nominal thickness of the plate, pipe, or tube before forming.
t_A measured average wall thickness of pipe or tube
t_B measured minimum wall thickness of the extrados of the bend.
t_f final thickness after forming.
θ acute angle of the spiral wrap measured from the longitudinal centerline.

6.10 Tables

Table 6.1 – Equations For Calculating Forming Strains

Type Of Part Being Formed	Forming Strain
For all **one piece** double curved circumferential products, formed by any process that includes dishing or cold spinning (for example, dished heads or cold spun heads),	$\varepsilon_f = 100 \ln\left(\dfrac{D_b}{D_0 - 2t}\right)$
Cylinders formed from plate:	$\varepsilon_f = \dfrac{50t}{R_f}\left(1 - \dfrac{R_f}{R_o}\right)$
For heads that are assembled from formed segments (for example, spherical dished shell plates or dished segments of elliptical or torispherical heads),	$\varepsilon_f = \dfrac{75t}{R_f}\left(1 - \dfrac{R_f}{R_o}\right)$
Tube and pipe bends:	$\varepsilon_f = \max\left[\left(\dfrac{r}{R_f}\right), \left(\dfrac{t_A - t_B}{t_A}\right)\right] \cdot 100$

Table 6.2 – Post Fabrication Strain Limits And Required Heat Treatment For High Alloy Materials (Note 1)

Grade	UNS Number	Limitations In Lower Temperature Range: For Design Temperature °C (°F) Exceeding	Limitations In Lower Temperature Range: For Design Temperature °C (°F) And less than or Equal to	Limitations In Lower Temperature Range: And Forming Strains Exceeding %	Limitations in Higher Temperature range: For Design Temperature Exceeding °C (°F)	Limitations in Higher Temperature range: And Forming Strains Exceeding %	Minimum Heat Treatment Temperature When Design Temperature Limits and Forming Strain Limits are Exceeded (Notes 3 and 4) °C (°F)
201-1	S20100 Heads	All	All	All	All	All	1065 (1950)
201-1	S20100 All Other	All	All	4	All	4	1065 (1950)
201-2	S20100 Heads	All	All	All	All	All	1065 (1950)
201-2	S20100 All Other	All	All	4	All	4	1065 (1950)
201LN	S20153 Heads	All	All	All	All	All	1065 (1950)
201LN	S20153 All Other	All	All	4	All	4	1065 (1950)
204	S20400 Heads	All	All	All	All	All	1065 (1950)
204	S20400 All Other	All	All	4	All	4	1065 (1950)
304	S30400	580 (1075)	675 (1250)	20	675 (1250)	10	1040 (1900)
304H	S30409	580 (1075)	675 (1250)	20	675 (1250)	10	1040 (1900)
304N	S30451	580 (1075)	675 (1250)	15	675 (1250)	10	1040 (1900)
309S	S30908	580 (1075)	675 (1250)	20	675 (1250)	10	1095 (2000)
310H	S31009	580 (1075)	675 (1250)	20	675 (1250)	10	1095 (2000)
310S	S31008	580 (1075)	675 (1250)	20	675 (1250)	10	1095 (2000)
316	S31600	580 (1075)	675 (1250)	20	675 (1250)	10	1040 (1900)
316H	S31609	580 (1075)	675 (1250)	20	675 (1250)	10	1040 (1900)
316N	S31651	580 (1075)	675 (1250)	15	675 (1250)	10	1040 (1900)
321	S32100	595 (1100)	675 (1250)	15 (note 5)	675 (1250)	10	1040 (1900)
321H	S32109	595 (1100)	675 (1250)	15 (note 5)	675 (1250)	10	1040 (2000)
347	S34700	595 (1100)	675 (1250)	15	675 (1250)	10	1040 (1900)
347H	S34709	595 (1100)	675 (1250)	15	675 (1250)	10	1095 (2000)
348	S34800	595 (1100)	675 (1250)	15	675 (1250)	10	1040 (1900)
348H	S34809	595 (1100)	675 (1250)	15	675 (1250)	10	1095 (2000)

Notes:

1. The limits shown are for cylinders formed from plates, spherical or dished heads formed from plate, and tube and pipe bends.
2. (Currently Not Used)
3. The rate of cooling from heat-treatment temperature is not subject to specific control limits.
4. While minimum heat-treatment temperatures are specified, it is recommended that the heat-treatment temperature range be limited to 85°C (150°F) above that minimum. The range can be extended to 140°C (250°F) above the maximum temperature range for 347, 347H, 348, and 348H).
5. For simple bends of tubes or pipes whose outside diameter is less than 90 mm (3 1/2 in.), this limit is 20%.

Table 6.3 – Post Fabrication Strain Limits And Required Heat Treatment For Nonferrous Materials (Note 1)

Grade	UNS Number	Limitations In Lower Temperature Range			Limitations in Higher Temperature range		Minimum Heat Treatment Temperature When Design Temperature Limits and Forming Strain Limits are Exceeded (Note2) °C (°F)
		For Design Temperature °C (°F)		And Forming Strains Exceeding %	For Design Temperature Exceeding °C (°F)	And Forming Strains Exceeding %	
		Exceeding	And less than or Equal to				
617	N06617	540(1000)	675 (1250)	15	675 (1250)	10	1150 (2100)
800	N08800	595 (1100)	675 (1250)	15	675 (1250)	10	985 (1800)
800H	N08810	595 (1100)	675 (1250)	15	675 (1250)	10	1120 (2050)
800HT	N08811	595 (1100)	675 (1250)	15	675 (1250)	10	1120 (2050)

Notes:

1. The limits shown are for cylinders formed from plates, spherical or dished heads formed from plate, and tube and pipe bends.
2. The rate of cooling from heat-treatment temperature is not subject to specific control limits.

Table 6.4 – Maximum Allowable Offset In Welded Joints

Section Thickness	Category A Joints	Category B, C, D Joints
13 mm (1/2 in) $\le t$	$t/4$	$t/4$
13 mm (1/2 in) $< t \le$ 19 mm (3/4 in)	3 mm (1/8 in)	$t/4$
19 mm (3/4 in) $< t \le$ 38 mm (1–1/2 in)	3 mm (1/8 in)	5 mm (3/16 in)
38 mm (1–1/2 in) $< t \le$ 50 mm (2 in)	3 mm (1/8 in)	$t/8$
$t >$ 50 mm (2 in)	$\max\left[t/16,\ 10\text{ mm (3/8 in)}\right]$	$\max\left[t/8,\ 19\text{ mm (3/4 in)}\right]$
Notes: t is the nominal thickness of the thinner section at the weld joint.		

Table 6.5 – Acceptable Welding Process And Limitations

Welding Process	Application/Limitation	Special Heat Treatment Requirement
• Gas metal arc • Gas tungsten arc • Plasma arc • Laser Beam	All material	None
• Electron beam	All material	Exception for post weld heat treatment as provided in paragraph 6.4.2 are not permitted when welding of ferritic materials greater than 3 mm (1/8 in.) in thickness.
• Shielded metal arc • Submerged arc • Explosive welding • Induction	All material except Titanium	None
• Electrogas • Electroslag	Butt weld only in ferritic steel and the following austenitic steels: • SA-240 – TP304, TP304L, TP316, TP316L • SA-182 – F304, F304L, TP316, TP316L, • SA-351 – CF3,CF3A, CF3M, CF8, CF8A , CF8M	For electro slag welding in ferritic materials over 38 mm (1 1/2 in.) in thickness at the joint or electrogas welding with a single pass greater than 38 mm (1 1/2 in.), the joint shall be given a grain refining (austenitizing) heat treatment.
• Inertia • Continuous drive friction	• Materials assigned a P-Number in Section IX excluding rimmed, semi-killed steel, or Titanium	Exceptions for post weld heat treatment as provided in paragraph 6.4.2 are not permitted when welding P-.No. 3, 4, 5A, 5B, 5C, 6, 7 (except TP405 and TP410S) and P.No.10
• Arc stud • Resistance stud	• Non-pressure parts having a load- or non-load-carrying function except for Quenched and Tempered High Strength Steels (see Table 3.A.4), provided that, in the case of ferrous materials, heat treatment requirements of paragraph 6.4.1 and 6.4.2 for the materials used in the vessel are met. • Stud shall be limited to 25 mm (1 in.) diameter for round studs and an equivalent cross section area for studs with other shapes.	In case of ferrous material, heat treatment requirements of paragraphs 6.4.3.6 and 6.6.6.3 for the materials used in the vessel shall be met.

Table 6.6 – Maximum Reinforcement For Welded Joints

Section Thickness	Circumferential Joints in Pipe and Tubing	Other Welds
2.5 mm (3/32 in) < t	2.5 mm (3/32 in.)	0.8 mm (1/32 in.)
2.5 mm (3/32 in) ≤ t < 5mm (3/16 in)	2.5 mm (3/32 in.)	1.5 mm (1/16 in.)
5 mm (3/16 in) ≤ t < 13 mm (1/2 in)	3 mm (1/8 in.)	2.5 mm (3/32 in.)
13 mm (1/2 in) ≤ t < 25 mm (1 in)	4.0 mm (5/32 in.)	2.5 mm (3/32 in.)
25 mm (1 in) ≤ t < 50 mm (2 in)	4.0 mm (5/32 in.)	3 mm (1/8 in.)
50 mm (2 in) ≤ t < 76 mm (3 in)	4.0 mm (5/32 in.)	4.0 mm (5/32 in.)
76 mm (3 in) ≤ t < 100 mm (4 in)	5.5 mm (7/32 in.)	5.5 mm (7/32 in.)
100 mm (4 in) ≤ t < 125 mm (5 in)	6 mm (1/4 in.)	6 mm (1/4 in.)
t ≥ 125mm (5 in)	8 mm (5/16 in.)	8 mm (5/16 in.)
Notes: t is the nominal thickness of the thinner section at the weld joint.		

Table 6.7 – Minimum Preheat Temperatures for Welding

P-No.	Minimum Preheat Temperature
1	80°C (175°F) for a material which has a specified maximum carbon content in excess of 0.30% and a thickness at the joint excess of 25 mm (1 in.) 10°C (50°F) for all other materials
3	80°C (175°F) for a material which has either a specified minimum tensile strength in excess of 480 MPa (70,000 psi) or a thickness at the joint in excess of 16 mm (5/8 in.) 10°C (50°F) for all other materials
4	120°C (250°F) for a material which has either a specified minimum tensile strength in excess of 410 MPa (60,000 psi) or a thickness at the joint in excess of 13 mm (1/2 in.) 10°C (50°F) for all other materials
5A, 5B, 5C	205°C (400°F) for a material which has either a specified minimum tensile strength in excess of 410 MPa (60,000 psi) or has both a specified minimum chromium content above 6.0% and a thickness at the joint in excess of 13 mm (1/2 in.) 150°C (300°F) for all other materials
6	205°C (400°F)
7	None
8	None
9A and 9B	150°C (300°F)
10A	150°C (300°F) with interpass temperature maintained between 175°C and 230°C (350°F and 450°F)
10F	120°C (250°F)
11A	for 5% and 9% Nickel steels preheat is neither required nor prohibited.
11B Gr. Gr. 1-6	80°C (175°F)
21 to 24, inclusive	None
31 to 35, inclusive	None
41 to 44, inclusive	None

Table 6.8 – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Material: P-No. 1, Group 1, 2, 3

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
a) PWHT is mandatory for the following conditions: 1) For welded joints over 38mm (1 1/2 in.) nominal thickness. 2) For welded joints over 32 mm (1 1/4 in.) through 38 mm (1 1/2 in.) nominal thickness unless a 95°C (200°F) minimum preheat is applied during welding b) When it is impractical to perform PWHT at the temperatures specified in this table, it is permissible to carry out PWHT at lower temperatures for longer periods of time in accordance with Table 6.16.	SI Units • For $t_n \le 50\ mm$: 595°C, 0.04 hr/mm, 15 minutes minimum • For: $50\ mm < t_n \le 125\ mm$ 595°C, 2 hr plus 0.6 minutes for each additional mm over 50 mm • For $t_n > 125\ mm$: 595°C, 2 hr plus 0.6minutes for each additional mm over 50 mm US Customary Units • For $t_n \le 2\ in$: 1100°F, 1 hr/in, 15 minutes minimum • For $2\ in < t_n \le 5\ in$: 1100°F, 2 hr plus 15 minutes for each additional inch over 2 in. • For $t_n > 5\ in$: 1100°F, 2 hr plus 15 minutes for each additional inch over 2 in.

Table 6.9 – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Material: P-No. 3, Group 1, 2, 3

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
a) PWHT is mandatory for P-No. 3, Gr. No.3 material in all thicknesses. b) PWHT is mandatory under the following conditions for other P number Group Number combinations: 1) On P-No 3, Gr. No. 1 and P-No. 3, Gr. No.2 material over 16 mm (5/8 in.) nominal thickness. For these materials, PWHT is mandatory on material up to and including 16 mm (5/8 in.) nominal thickness unless a welding procedure qualification described in paragraph 6.2.2.4 has been made for equal or greater thickness than the production weld. 2) If for pressure parts subject to direct firing. c) For welding connections and attachments to pressure parts, PWHT is not mandatory under the conditions specified below: 1) For attaching to pressure parts which have a specified maximum carbon content of not more than 0.25% (SA Material Specification carbon content, except when further limited by the Purchaser to a value within the Specification limits) or to non-pressure parts with groove welds not over 13 mm (1/2 in.) size or fillet welds having a throat thickness of 13 mm (1/2 in.). or. less, provided preheat to a minimum temperature of 95°C (200°F) is applied; 2) For circumferential butt welds in pipe or tube where the pipe or tube has both a nominal wall thickness of 13 mm (1/2 in.) or less and a specified maximum carbon content of not more than 0.25% (SA Material Specification carbon content, except when further limited by the Purchaser to a value within the Specification limits); 3) For studs welded to pressure parts which have a specified maximum carbon content of not more than 0.25% (SA Material Specification carbon content, except when further limited by the Purchaser to a value within the Specification limits), provided preheat to a minimum temperature of 95°C (200°F) is applied; 4) For corrosion resistant weld metal overlay cladding or for welds attaching corrosion resistant applied lining when welded to pressure parts which have a specified maximum carbon content of not more than 0.25% (SA Material Specification carbon content, except when further limited by the Purchaser to a value within the Specification limits), provided preheat to a minimum temperature of 95°C (200°F) is; maintained during application of the first layer. d) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.4.3, additional test coupons shall be made and tested. e) When it is impractical to perform PWHT at the temperatures specified in this table, it is permissible to carry out PWHT at lower temperatures for longer periods of time in accordance with Table 6.16. When PWHT is performed in accordance with this provision, the vessel test plate required by paragraph 6.5.4 shall receive the same heat treatment.	SI Units • For $t_n \le 50\ mm$: 595°C, 0.04 hr/mm, 15 minutes minimum • For: $50\ mm < t_n \le 125\ mm$ 595°C, 2 hr plus 0.6 minutes for each additional mm over 50 mm • For $t_n > 125\ mm$: 595°C, 2 hr plus 0.6 minutes for each additional mm over 50 mm US Customary Units • For $t_n \le 2\ in$: 1100°F, 1 hr/in, 15 minutes minimum • For $2\ in < t_n \le 5\ in$: 1100°F, 2 hr plus 15 minutes for each additional inch over 2 in. • For $t_n > 5\ in$: 1100°F, 2 hr plus 15 minutes for each additional inch over 2 in.

Table 6.10 – Requirements for Post Weld Heat Treatment (PWHT) of Pressure Parts and Attachments For Materials: P-No. 4, Group 1, 2

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
a) PWHT is mandatory under the following conditions: 1) On material of SA-202 Grades A and B over 16 mm (5/8 in.) nominal thickness. For these materials, PWHT is mandatory up to and including 16 mm (5/8 in.) nominal thickness unless a Welding Procedure Qualification described in paragraph 6.2.2.4 has been made in equal or greater thickness than the production weld. 2) on material of all thicknesses for pressure parts subject to direct firing 3) On all other P. No. 4 Gr. Nos. 1 and 2 materials. b) PWHT is not mandatory under the conditions specified below: 1) For circumferential butt welds in pipe or tube of P-No. 4 materials where the pipe or tubes comply with all of the following conditions: i) a maximum nominal outside diameter of 100 mm (4 in.); ii) a maximum nominal thickness of 16 mm (5/8 in.).; iii) maximum specified carbon content of not more than 0.15% (SA material Specification carbon content, except when further limited by the Purchaser to a value within the Specification limits); iv) a minimum preheat of 120°C (250°F) . 2) For P-No. 4 pipe or tube materials meeting the requirements of (a)(1), (a)(2), and (a)(3) above, having nonpressure attachments fillet welded to them provided; i) the fillet welds have a maximum throat thickness of 13 mm (1/2 in.); ii) a minimum preheat temperature of 120°C (250°F) is applied. 3) For P-No. 4 pipe or tube materials meeting the requirements of (a)(1), (a)(2), and (a)(3) above, having studs welded to them provided a minimum preheat temperature of 120°C (250°F) is applied. c) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.4.3, additional test coupons shall be made and tested.	SI Units • For $t_n \le 50\ mm$: 650°C, 0.04 hr/mm, 1 hr minimum • For: $50\ mm < t_n \le 125\ mm$ 650°C, 0.04 hrs/mm • For $t_n > 125\ mm$: 650°C, 5 hr plus 0.6 minutes for each additional mm over 125 mm US Customary Units • For $t_n \le 2\ in$: 1200°F, 1 hr/in, 1 hr minimum • For $2\ in < t_n \le 5\ in$: 1200°F, 1 hr/in. • For $t_n > 5\ in$: 1200°F, 5 hr plus 15 minutes for each additional inch over 5 in.

Table 6.11 – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Materials: P-No. 5A, P-No. 5B Group 1, and P-No. 5C Group 1

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
a) Except under the following conditions, PWHT is mandatory: 1) For circumferential butt welds in pipe or tubes where the pipe or tubes comply with all of the following conditions: i) a maximum specified chromium content of 3.0%; ii) a maximum nominal outside diameter of 100 mm (4 in.); iii) a maximum nominal thickness of 16 mm (5/8 in.); iv) a maximum specified carbon content of not more than 0.15% (SA Material Specification carbon content, except when further limited by the Purchaser to a value within the Specification limits); v) a minimum preheat of 150°C (300°F) is applied. 2) For pipe *or* tube materials meeting the requirements of (1)(i), (1)(ii), (1)(iii), and (1)(iv) having nonpressure attachments fillet welded to them provided: i) the fillet welds have a maximum throat thickness of 13mm (1/2 in.); ii) a minimum preheat temperature of 150°C (300°F) is applied. 3) For pipe or tube materials meeting the requirements of (1)(i), (1)(ii), (1)(iii), and {1)(iv) having studs welded to them provided a minimum preheat temperature of 150°C (300°F) is applied. b) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.4.3, additional test coupons shall be made and tested. c) When it is impractical to postweld heat treat P-No. 5A, 5B Group No. 1, and 5C Group No.1 materials at the temperature specified in this Table, it is permissible to perform the PWHT at 650°C (1200°F) minimum provided that, for material up to 50 mm (2 in.) nominal thickness, the holding time is increased to the greater of 4 hr minimum or 9.6 min/mm (4 hr/in.). of thickness; for thickness over 50 mm (2 in.), the specified holding times are multiplied by 4. The requirements in paragraph 3.4.3 must be accommodated in this reduction in PWHT.	SI Units P-No. 5A, P-No. 5B Group 1, and P-No. 5C Group 1 • For $t_n \le 50\ mm$: 675°C, 0.04 hr/mm, 1 hour minimum • For $50\ mm < t_n \le 125\ mm$: 675°C, 0.04 hr/mm • For $t_n > 125\ mm$: 675°C, 5 hr plus 0.6 minutes for each additional mm over 125 mm US Customary Units P-No. 5A, P-No. 5B Group 1, and P-No. 5C Group 1 • For $t_n \le 2\ in$: 1250°F, 1 hr/in, 1 hour minimum • For $2\ in < t_n \le 5\ in$: 1250°F, 1 hr/in • For $t_n > 5\ in$: 1250°F, 5 hr plus 15 minutes for each additional inch over 5 in

Table 6.11.A – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Materials: P-No. 15E Group 1

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
a) If the nominal thickness is ≤13 mm (1/2 in.), the minimum holding temperature is 720 °C (1325°F). b) For dissimilar metal welds (i.e., welds made between a P-No.15 E Group 1 and another lower chromium ferritic, austenitic, or nickel-based steel), if the filler metal chromium content is less than 3.0% or if the filler metal is nickel-based or austenitic, the minimum holding temperature shall be 705°C (1300°F). c) The maximum holding temperature above is to be used if the actual chemical composition of the matching filler metal used when making the weld is unknown. If the chemical composition of the matching filler metal is known, the maximum holding temperature can be increased as follows: . 1) If Ni + Mn <1.50% but ≥ 1.0%, the maximum PWHT temperature is 790˚C (1450°F) 2) If Ni + Mn < 1.0% the maximum PWHT temperature is 800˚ C (1470°F) 3) The lower transformation temperature for matching filler material is affected by alloy content, primarily the total Ni + Mn. The maximum holding temperature has been set to avoid heat treatment in the intercritical zone. d) If a portion of the component is heated above the heat treatment temperature allowed above, one of the following actions shall be performed: 1) The component in its entirety must be renormalized and tempered. 2) If the maximum holding temperature in the Table or (c)(1) is exceeded, but does not exceed 800˚ C (1470˚ F), the weld metal shall be removed and replaced. 3) The portion of the component heated above 800˚C (1470˚F) and at least 75 mm (3in.) on either side of the overheated zone must be removed and be renormalized and tempered, or replaced. 4) The allowable stress shall be that for Grade 9 material (i.e., SA-213-T9, SA- 335- P9, or equivalent product specification) at the design temperature, provided that the portion of the component heated to a temperature greater than the allowed above is reheat treated within the temperature range specified above. e) Postweld heat treatment is not mandatory for electric resistance welds used to attach extended heat -absorbing fins to pipe and tube materials provided the following requirements are met: 1) a maximum pipe or tube size of 100 DN (NPS 4) 2) a maximum specified carbon content (SA material specification carbon content, except when further limited by the Purchaser to a value within the specification limits) of not more than 0.15 % 3) a maximum fin thickness of 3 mm (1/8 in.) 4) prior to using the welding procedure, the Manufacturer shall demonstrate that the heat-affected zone does not encroach upon the minimum wall thickness.	<u>SI Units</u> <u>P-No. 15E Group 1</u> • For $t \le 125 mm$: 730°C min., 775˚ C max. 0.04 hr/mm, 30 minutes minimum • For $t_n > 125$ *mm* : 730°C min. 775˚ C max., 5 hr plus 0.6 minutes for each additional mm over 125 mm <u>US Customary Units</u> <u>P-No. 15E Group 1</u> • For t≤ 5 inch: 1350°F min. 1425˚ F max. , 1 hr/in, 30 minutes minimum. • For $t_n > 5$ *in* : 1350°F min, 1425˚ F max., 5 hr plus 15 minutes for each additional inch over 5 inches.

Table 6.12 – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Materials: P-No. 6, Group 1, 2, 3

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
a) PWHT is not required for vessels constructed of Type 410 material with carbon content not to exceed 0.08% and welded with electrodes that produce an austenitic chromium-nickel weld deposit or a non air-hardening nickel-chromium iron weld deposit, provided the plate thickness at the welded joint does not exceed 10 mm (3/8 in.), and for thicknesses over 10 mm (3/8 in.). to 38 mm (1 ½ in.) provided a preheat of 230°C (450°F) is maintained during welding and that the joints are completely radiographed. b) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.4.3, additional test coupons shall be made and tested.	SI Units • For $t_n \le 50\ mm$: 760°C, 0.04 hr/mm, 1 hr minimum • For: $50\ mm < t_n \le 125\ mm$ 760°C, 2 hr plus 0.6 minutes for each additional mm over 50 mm • For $t_n > 125\ mm$: 760°C, 2 hr plus 0.6 minutes for each additional mm over 50 mm US Customary Units • For $t_n \le 2\ in$: 1400°F, 1 hr/in, 1 hr minimum • For $2\ in < t_n \le 5\ in$: 1400°F, 2 hr plus 15 minutes for each additional inch over 2 in. • For $t_n > 5\ in$: 1400°F, 2 hr plus 15 minutes for each additional inch over 2 in.

Table 6.13 – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Materials: P-No. 7, Group 1, 2 and P-No. 8

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
Materials: P-No. 7, Group 1, 2	
a) PWHT shall be performed as prescribed in paragraph 6.2.6 except that the cooling rate shall be a maximum of 55°C (100°F) per hour in the range above 650°C (1200°F), after which the cooling rate shall be sufficiently rapid to prevent embrittlement. PWHT is not required for vessels constructed of Type 405 and Type 410S material with carbon content not to exceed 0.08%, welded with electrodes that produce an austenitic chromium-nickel weld deposit or a non air-hardening nickel-chromium-iron weld deposit, provided the plate thickness at the welded joint does not exceed 3mm (1/8 in.), and for thicknesses over 3 mm (1/8 in.) to 38 mm (1 1/2 in.) provided a preheat of 230°C (450°F) is maintained during welding and that the joints are completely radiographed. b) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.4.3, additional test coupons shall be made and tested.	SI Units • For $t_n \le 50\ mm$: 730°C, 0.04 hr/mm, 1 hr minimum • For: $50\ mm < t_n \le 125\ mm$ 730°C, 2 hr plus 0.6 minutes for each additional mm over 50 mm • For $t_n > 125\ mm$: 730°C, 2 hr plus 0.6 minutes for each additional mm over 50 mm US Customary Units • For $t_n \le 2\ in$: 1350°F, 1 hr/in, 1 hr minimum • For $2\ in < t_n \le 5\ in$: 1350°F, 2 hr plus 15 minutes for each additional inch over 2 in. • For $t_n > 5\ in$: 1350°F, 2 hr plus 15 minutes for each additional inch over 2 in.
Materials: P-No. 8	
PWHT is neither required nor prohibited.	

Table 6.14 – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Materials: P-No. 9A, Group 1 and P-No. 9B, Group 1

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
Materials: P-No. 9A, Group 1	
a) PWHT is mandatory under the following conditions: 1) On material of all thicknesses if required by the Purchasers Design Specification 2) On material over 16 mm (5/8 in.) nominal thickness. For material up to and including 16 mm (5/8 in.) nominal thickness, postweld heat treatment is mandatory unless a welding procedure qualification described in paragraph 6.2.1.1 has been made in equal or greater thickness than the production weld. 3) or if for pressure parts subject to direct firing. b) PWHT is not mandatory under the conditions specified below: 1) for circumferential butt welds in pipe or tubes where the pipe or tubes comply with all of the following conditions: i) a maximum nominal outside diameter of 100 mm (4 in.); ii) a maximum thickness of 13 mm (1/2 in.); iii) a maximum specified carbon content of not more than 0.15% (SA Material Specification carbon content, except when further limited by the Purchaser to a value within the Specification limits); iv) a minimum preheat of 120°C (250°F). 2) for pipe or tube materials meeting the requirements of (1)(i), (1)(ii), and (1)(iii) above, having attachments fillet welded to them, provided; i) the fillet welds have a throat thickness of 13 mm (1/2 in.), or less; ii) the material is preheated to 120°C (250°F) minimum. A lower preheating temperature may be used provided specifically controlled procedures necessary to produce sound welded joints are used. Such procedures shall include but not be limited to the following: • the throat thickness of fillet welds shall be 13 mm (1/2 in.) or less; • the maximum continuous length of fillet welds shall be not over 100 mm (4 in.); • the thickness of the test plate used in making the welding procedure qualification of Section IX shall not be less than that of the material to be welded. 3) for attaching nonpressure parts to pressure parts with groove welds not over 13 mm (1/2 in.) in size or fillet welds that have a throat thickness of 13 mm (1/2in.) or less, provided preheat to a minimum temperature of 95°C (200°F) is applied; 4) for studs welded to pressure parts provided preheat to a minimum temperature of 95°C (200°F) is applied; 5) for corrosion resistant weld metal overlay cladding or for welds attaching corrosion resistant applied lining provided preheat to a minimum temperature of 95°C (200°F) is maintained during application of the first layer. c) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.4.3, additional test coupons shall be made and tested.	SI Units 595°C: 1 hr minimum, plus 0.6 min/mm for thickness over 25 mm US Customary Units 1100°F: 1 hr minimum, plus 15 min/in. for thickness over 1 in.

Table 6.14 – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Materials: P-No. 9A, Group 1 and P-No. 9B, Group 1

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
d) When the heating rate is less than 30°C/hr (50°F/hr) between 430°C (800°F) and the holding temperature, the additional 0.60 min/mm (15 min/in.) holding time is not required. Additionally, where the Manufacturer can provide evidence that the minimum temperature has been achieved throughout the thickness, the additional 0.60 min/mm (15 min/in.) holding time is not required. e) When it is impractical to postweld heat treat at the temperature specified in this Table, it is permissible to carry out the PWHT at lower temperatures 540°C (1000°F minimum)) for longer periods of time in accordance with Table 6.16. When PWHT is performed in accordance with this provision, the vessel test plate required by paragraph 3.4.3 shall receive the same heat treatment.	
Materials: P-No. 9B, Group 1	
a) PWHT is mandatory under the following conditions: 1) On material over 16 mm (5/8 in.) nominal thickness. For material up to and including 16 mm (5/8 in.) nominal thickness, PWHT is mandatory unless a welding procedure qualification described in paragraph 6.2.2.4 has been made in equal or greater thickness than the production weld. 2) On pressure parts subject to direct firing b) PWHT is not mandatory under the conditions specified below: 1) For attaching nonpressure parts to pressure parts with groove welds not over 13 mm (1/2 in.) in size or fillet welds that have a throat thickness of 13 mm (1/2in.) or less, provided preheat to a minimum temperature of 95°C (200°F) is applied; 2) For studs welded to pressure parts provided preheat to a minimum temperature of 95°C (200°F) is applied; 3) For corrosion resistant weld metal overlay cladding or for welds attaching corrosion resistant applied lining (see 6.5.5.1), provided preheat to a minimum temperature of 95°C (200°F) is maintained during application of the first layer. c) The holding temperature for PWHT shall not exceed 635°C (1175°F). d) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.4.3, additional test coupons shall be made and tested. e) When the heating rate is less than 30°C/hr (50°F/hr) between 430°C (800°F) and the holding temperature, the additional 0.6 min/mm (15 min/in.) holding time is not required. Additionally, where the Manufacturer can provide evidence that the minimum temperature has been achieved throughout the thickness, the additional 0.6 min/mm (15 min/in.) holding time is not required. f) When it is impractical to postweld heat treat at the temperature specified in this Table, it is permissible to carry out the PWHT at lower temperatures 540°C (1000°F) minimum for longer periods of time in accordance with Table 6.16. When PWHT is performed in accordance with this provision, the vessel test plate required by paragraph 3.4.3 shall receive the same heat treatment.	<u>SI Units</u> 595°C: 1 hr minimum, plus 0.6 min/mm for thickness over 25 mm <u>US Customary Units</u> 1100°F: 1 hr minimum, plus 15 min/in. for thickness over 1 in.

Table 6.15 – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Materials: P-No. 10A, Group 1; P-No. 10B, Group 2; P-No. 10C, Group 1, P-No. 10E, Group 1; P-No. 10F, Group 6; P-No. 10G, Group 1; P-No. 10H, Group 1; P-No. 10I, Group 1; and P-No. 10K, Group 1

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
Materials: P-No. 10A, Group 1	
a) Postweld heat treatment is mandatory under the following conditions: 1) On all thicknesses of SA-487 Cl. 1A material; 2) On all other P-No. 10A materials over 16 mm (5/8 in.) nominal thickness. For these materials, postweld heat treatment is mandatory on material up to and including 16 mm (5/8 in.) nominal thickness, unless a welding procedure qualification described in paragraph 6.2.2.4 has been made in equal or greater thickness than the production weld. 3) On pressure parts subject to direct firing. b) Postweld heat treatment is not mandatory under the conditions specified below: 1) For attaching to pressure parts which have a specified maximum carbon content of not more than 0.25%. (SA Material Specification carbon content, except when further limited by the Purchaser to a value within the Specification limits) or to nonpressure parts with groove welds not over 13 mm (1/2 in.) in size or fillet welds having a throat thickness of 13 mm (1/2 in.) or less, provided preheat to a minimum temperature of 95°C (200°F) is applied; 2) For circumferential butt welds in pipes or tube where the pipe or tube has both a nominal wall thickness of 13 mm (1/2 in.) or less and a specified maximum carbon content of not more than 0.25% (SA Material Specification carbon content, except when further limited by the Purchaser to a value within the Specification limits), provided preheat to a minimum temperature of 95°C (200°F) is applied; 3) For studs welded to pressure parts which have a specified maximum carbon content of not more than 0.25 % (SA Material Specification carbon content, except when further limited by the Purchaser to a value within the Specification limits), provided preheat to a minimum temperature of 95°C (200°F) is applied; 4) For corrosion resistant weld metal overlay cladding or for welds attaching corrosion resistant applied lining (see 6.5.5.1) when welded to pressure parts which have a specified maximum carbon content of not more than 0.25% (SA Material Specification carbon content, except when further limited by the Purchaser to a value within the Specification limits), provided preheat to a minimum temperature of 95°C (200°F) maintained during application of the first layer. c) Consideration should be given for possible embrittlement of materials containing up to 0.15% vanadium when postweld heat treatment is performed at the minimum temperature and at lower temperatures for longer holding times. d) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.11.8.4, additional test coupons shall be made and tested. e) When the heating rate is less than 30°C/hr (50°F/hr) between 430°C (800°F) and the holding temperature, the additional 0.6 min/mm (15 min/in.) holding time is not required. Additionally, where the Manufacturer can provide evidence that the minimum temperature has been achieved throughout the thickness, the additional 0.6 min/mm (15 min/in.) holding time is not required.	<u>SI Units</u> 595°C: 1 hr minimum, plus 0.6 min/mm for thickness over 25 mm <u>US Customary Units</u> 1100°F: 1 hr minimum, plus 15 min/in. for thickness over 1 in.

Table 6.15 – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Materials: P-No. 10A, Group 1; P-No. 10B, Group 2; P-No. 10C, Group 1, P-No. 10E, Group 1; P-No. 10F, Group 6; P-No. 10G, Group 1; P-No. 10H, Group 1; P-No. 10I, Group 1; and P-No. 10K, Group 1

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
f) When it is impractical to postweld heat treat at the temperature specified in this Table, it is permissible to carry out the postweld heat treatment at lower temperatures for longer periods of time in accordance with Table 6.16. When postweld heat treatment is performed in accordance with this Note, the vessel test plate required by paragraph 3.11.8.4 shall receive the same heat treatment.	
Materials: P-No. 10B, Group 2	
a) Postweld heat treatment is mandatory for P-No. 10B materials for all thicknesses. b) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.11.8.4, additional test coupons shall be made and tested. c) When the heating rate is less than 30°C/hr (50°F /hr) between 430°C (800°F) and the holding temperature, the additional 0.6 min/mm (15 min/in.) holding time is not required. Additionally, where the Manufacturer can provide evidence that the minimum temperature has been achieved throughout the thickness, the additional 0.6 min/mm (15 min/in.) holding time is not required.	<u>SI Units</u> 595°C: 1 hr minimum, plus 0.6 min/mm for thickness over 25 mm <u>US Customary Units</u> 1100°F: 1 hr minimum, plus 15 min/in. for thickness over 1 in.
Materials: P-No. 10C, Group 1	
a) Postweld heat treatment is mandatory under the following conditions: 1) For welded joints over 38mm (1 1/2 in.) nominal thickness. 2) For welded joints over 32 mm (1 1/4 in.) through 38 mm (1 1/2 in.) nominal thickness unless a 95°C (200°F) minimum preheat is applied during welding b) Postweld heat treatment is not mandatory under the conditions specified below: 1) For groove welds not over 13 mm (1/2 in.) in size and fillet welds with throat not over 13 mm (1/2 in.) that attach nozzle connections that have a finished inside diameter not greater than 50 mm (2 in.), provided the connections do not form ligaments that require an increase in shell or head thickness and preheat to a minimum temperature of 95°C (200°F) is applied; 2) For groove welds not over 13 mm (1/2 in.) in size or fillet welds having throat thickness of 13 mm (1/2 in.) or less used for attaching nonpressure parts to pressure parts and preheat to a minimum temperature of 95°C (200°F) is applied when the thickness of the pressure part exceeds 32 mm (1 ¼ in.); 3) For studs welded to pressure parts provided preheat to a minimum temperature of 95°C (200°F) is applied when the thickness of the pressure part exceeds 32 mm (1 ¼ in.); 4) For corrosion resistant weld metal overlay cladding or for welds attaching corrosion resistant applied lining (see 6.5.5.1), provided preheat to a minimum temperature of 95°C (200°F) is maintained during application of the first layer when the	<u>SI Units</u> 540°C: 1 hr minimum, plus 0.6 min/mm for thickness over 25 mm <u>US Customary Units</u> 1000°F: 1 hr minimum, plus 15 min/in. for thickness over 1 in.

Table 6.15 – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Materials: P-No. 10A, Group 1; P-No. 10B, Group 2; P-No. 10C, Group 1, P-No. 10E, Group 1; P-No. 10F, Group 6; P-No. 10G, Group 1; P-No. 10H, Group 1; P-No. 10I, Group 1; and P-No. 10K, Group 1

PWHT Requirements	Holding Temperature and Time Based On The Nominal Thickness
thickness of the pressure part exceeds 32 mm (1 ¼ in.). c) When the heating rate is less than 30°C/hr (50°F/hr) between 430°C (800°F) and the holding temperature, the additional 0.6 min/mm (15 min/in.) holding time is not required. Additionally, where the Manufacturer can provide evidence that the minimum temperature has been achieved throughout the thickness, the additional 0.6 min/mm (15 min/in.) holding time is not required. d) When it is impractical to postweld heat treat at the temperature specified in this Table, it is permissible to carry out the postweld heat treatment at lower temperatures for longer periods of time in accordance with Table 6.16.	
Materials: P-No. 10E, Group 1	
a) For SA-268 Grade TP446 material only, the cooling rate shall be a maximum of 55°C/hr (100°F/hr) in the range above 650°C (1200°F) after which the cooling rate shall be sufficiently rapid to prevent embrittlement. b) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.11.8.4 additional test coupons shall be made and tested. c) When the heating rate is less than 30°C/hr (50°F/hr) between 430°C (800°F) and the holding temperature, the additional 0.6 min/mm (15 min/in.) holding time is not required. Additionally, where the Manufacturer can provide evidence that the minimum temperature has been achieved throughout the thickness, the additional 0.6 min/mm (15 min/in.) holding time is not required.	<u>SI Units</u> 675°C: 1 hr minimum, plus 0.6 min/mm for thickness over 25 mm <u>US Customary Units</u> 1250°F: 1 hr minimum, plus 15 min/in. for thickness over 1 in.
Materials: P-No. 10F, Group 6	
a) Postweld heat treatment is mandatory for P-No. 10F materials for all thicknesses. b) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.11.8.4, additional test coupons shall be made and tested. c) When the heating rate is less than 30°C/hr (50°F/hr) between 430°C (800°F) and the holding temperature, the additional 0.6 min/mm (15 min/in.) holding time is not required. Additionally, where the Manufacturer can provide evidence that the minimum temperature has been achieved throughout the thickness, the additional 0.6 min/mm (15 min/in.) holding time is not required.	<u>SI Units</u> 595°C: 1 hr minimum, plus 0.6 min/mm for thickness over 25 mm <u>US Customary Units</u> 1100°F: 1 hr minimum, plus 15 min/in. for thickness over 1 in.
Materials: P-No. 10G, Group 1	
a) PWHT is neither required nor prohibited. b) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.11.8.4, additional test coupons shall be made and tested.	

Table 6.15 – Requirements For Post Weld Heat Treatment (PWHT) Of Pressure Parts And Attachments For Materials: P-No. 10A, Group 1; P-No. 10B, Group 2; P-No. 10C, Group 1, P-No. 10E, Group 1; P-No. 10F, Group 6; P-No. 10G, Group 1; P-No. 10H, Group 1; P-No. 10I, Group 1; and P-No. 10K, Group 1

<table>
<tr><th>PWHT Requirements</th><th>Holding Temperature and Time Based On The Nominal Thickness</th></tr>
<tr><td colspan="2">Materials: P-No. 10H, Group 1</td></tr>
<tr><td>For the austenitic-ferritic wrought or cast duplex stainless steels listed below, PWHT is neither required nor prohibited. However, if heat treatment is performed, it shall be performed as listed below and followed by liquid quenching or rapid cooling by other means:
<table>
<tr><th rowspan="2">Alloy</th><th colspan="2">PWHT Temperature</th></tr>
<tr><th>°C</th><th>°F</th></tr>
<tr><td>S32550</td><td>1030 – 1120</td><td>1900 – 2050</td></tr>
<tr><td>S31803</td><td>1020 – 1100</td><td>1870 – 2010</td></tr>
<tr><td>S32900 (0.08 max. C)</td><td>940 – 955</td><td>1725 – 1750</td></tr>
<tr><td>S31200</td><td>1040 – 1095</td><td>1900 – 2000</td></tr>
<tr><td>S31500</td><td>975 – 1025</td><td>1785 – 1875</td></tr>
<tr><td>S32404</td><td>950 – 1050</td><td>1740 – 1920</td></tr>
<tr><td>J93345</td><td>1120 minimum</td><td>2050 minimum</td></tr>
<tr><td>S32750</td><td>980 –1125</td><td>1800 –2060</td></tr>
<tr><td>S32950</td><td>995 – 1025</td><td>1825 – 1875</td></tr>
</table></td><td></td></tr>
<tr><td colspan="2">Materials: P-No. 10I, Group 1</td></tr>
<tr><td>a) The cooling rate shall be a maximum of 55°C/hr (100°F/hr) in the range above 650°C (1200°F) after which the cooling rate shall be rapid to prevent embrittlement.
b) PWHT is neither required nor prohibited for a thickness of 13 mm (1/2 in.) or less.
c) For alloy S44635, the rules for ferritic chromium stainless steel shall apply, except that PWHT is neither prohibited nor required. If heat treatment is performed after forming or welding, it shall be performed at 1010°C (1850°F) minimum followed by rapid cooling to below 430°C (800°F).
d) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.11.8.4, additional test coupons shall be made and tested.
e) When the heating rate is less than 30°C/hr (50°F/hr) between 430°C (800°F) and the holding temperature, the additional 0.6 min/mm (15 min/in.) holding time is not required. Additionally, where the Manufacturer can provide evidence that the minimum temperature has been achieved throughout the thickness, the additional 0.6 min/mm (15 min/in.) holding time is not required.</td><td><u>SI Units</u>
730°C: 1 hr minimum, plus 0.6 min/mm for thickness over 25 mm

<u>US Customary Units</u>
1350°F: 1 hr minimum, plus 15 min/in. for thickness over 1 in.</td></tr>
<tr><td colspan="2">Materials: P-No. 10K, Group 1</td></tr>
<tr><td>a) For alloy S44660, the rules for ferritic chromium stainless steel shall apply, except that PWHT is neither required nor prohibited. If heat treatment is performed after forming or welding, it shall be performed at 815°C (1500°F) to 1065°C (1950°F) for a period not to exceed 10 min followed by rapid cooling.
b) If during the holding period of PWHT, the maximum time or temperature of any vessel component exceeds the provisions of paragraph 3.11.8.4, additional test coupons shall be made and tested.</td><td></td></tr>
</table>

Table 6.16 – Alternative Postweld Heat Treatment Requirements (Applicable Only When Permitted by Tables 6.8 through 6.15)

Decrease in Temperature Below Minimum Specified Temperature, °C (°F)	Minimum Holding Time at Decreased Temperature, hr (Note 1)
30 (50)	2
55 (100)	4
85 (150) (Note 2)	10
110 (200) (Note 2)	20

Notes:

1. Minimum holding time for 25 mm (1 in.) thickness or less. Add 0.6 min/mm (15 min/in.) of thickness for thicknesses greater than 25 mm (1 in.)
2. These lower Postweld Heat Treat temperatures are permitted only for P-No 1, Group 1 and Group 2 materials.

Table 6.17 – Postweld Heat Treatment Requirements For Quenched And Tempered Materials In Part 3, Table 3.A.4

Specification	Grade or Type	P-No. and Group No.	(Nominal) Thickness Requiring PWHT mm (in.)	Postweld Heat Treatment Temp. °C (°F)	Holding Time hr/25 mm (hr/in)	Minimum Holding Time hr
Plate Steels						
SA-353	9Ni	11A Gr. 1	Over 50 (2)	550 - 585 (1025-1085)	1	2
SA-517	Grade A	11B Gr. 1	Over 15 (0.58)	540 - 595 (1000-1100)	1	1/4
SA-517	Grade B	11B Gr. 4	Over 15 (0.58)	540 - 595 (1000-1100)	1	1/4
SA-517	Grade E	11B Gr. 2	Over 15 (0.58)	540 - 595 (1000-1100)	1	1/4
SA-517	Grade F	11B Gr. 3	Over 15 (0.58)	540 - 595 (1000-1100)	1	1/4
SA-517	Grade J	11B Gr. 6	Over 15 (0.58)	540 - 595 (1000-1100)	1	1/4
SA-517	Grade P	11B Gr. 8	Over 15 (0.58)	540 - 595 (1000-1100)	1	1/4
SA-533	Grades A, B, C & D, Cl. 2	3 Gr. 3	All	540 - 565 (1000-1050)	½	1/2
SA-533	Grades B & D, Cl. 3	11A Gr. 4	Over 15 (0.58)	540 - 565 (1000-1050)	½	1/2
SA-543	Types B & C, Cl. 1	11A Gr. 5	Note 2	540 - 565 (1000-1050)	1	1
SA-543	Types B & C, Cl. 2	11A Gr. 5	Note 2	540 - 565 (1000-1050)	1	1
SA-543	Types B & C, Cl. 3	11A Gr. 5	Note 2	540 - 565 (1000-1050)	1	1
SA-553	Types I & II	11A Gr. 1	Over 50 (2)	550 - 585 (1025-1085)	1	2
SA-645	Grade A	11A Gr. 2	Over 50 (2)	550 - 585 (1025-1085)	1	2
SA-724	Grades A & B	1 Gr. 4	None (Note 3)	Note 1	Note 1	Note 1
SA-724	Grades A & B	1 Gr. 4	Over 22 (7/8) (Note 3)	565-620 (1050-1150)		
SA-724	Grade C	1 Gr. 4	Over 38 (1 1/2)	565 -620 (1050-1150)	1	1/2
Pipes and Tubes						
SA-333	Grade 8	11A Gr. 1	Over 50 (2)	550 - 585 (1025-1085)	1	2
SA-334	Grade 8	11A Gr. 1	Over 50 (2)	550 - 585 (1025-1085)	1	2
Forgings						
SA-372	Grade D		See 6.7.6.3 and SA-372 for heat treating requirements			
SA-372	Grade E, Cl. 70		See 6.7.6.3 and SA-372 for heat treating requirements			

Table 6.17 – Postweld Heat Treatment Requirements For Quenched And Tempered Materials In Part 3, Table 3.A.4

Specification	Grade or Type	P-No. and Group No.	(Nominal) Thickness Requiring PWHT mm (in.)	Postweld Heat Treatment Temp. °C (°F)	Holding Time hr/25 mm (hr/in)	Minimum Holding Time hr
SA-372	Grade F, Cl. 70		See 6.7.6.3 and SA-372 for heat treating requirements			
SA-372	Grade G, Cl. 70		See 6.7.6.3 and SA-372 for heat treating requirements			
SA-372	Grade H, Cl. 70		See 6.7.6.3 and SA-372 for heat treating requirements			
SA-372	Grade J, Cl. 70		See 6.7.6.3 and SA-372 for heat treating requirements			
SA-372	Grade J, Cl. 110		See 6.7.6.3 and SA-372 for heat treating requirements			
SA-508	Grade 4N, Cl. 1	11A Gr. 5	Note 2	540 - 565 (1000-1050)	1	1
SA-508	Grade 4N, Cl. 2	11A Gr. 5	Note 2	540 - 565 (1000-1050)	1	1
SA-522	Type 1	11A Gr. 1	Over 50 (2)	550 - 585 (1025-1085)	1	2
SA-592	Grade A	11B Gr. 1	Over 15 (0.58)	540 - 595 (1000-1100)	1	1/4
SA-592	Grade E	11B Gr. 2	Over 15 (0.58)	540 - 595 (1000-1100)	1	1/4
SA-592	Grade F	11B Gr. 3	Over 15 (0.58)	540 - 595 (1000-1100)	1	1/4

Notes:
1. NA indicates not applicable.
2. PWHT is neither required nor prohibited. Consideration should be given to the possibility of temper embrittlement. The cooling rate from PWHT, when used, shall not be slower than that obtained by cooling in still air.
3. PWHT required for thickness above 22 mm (7/8 in.).

Table 6.18 –Quench And Tempered Steels Conditionally Exempt From Production Impact Tests

Specification	UNS	P-No./Group Number
SA-353	K81340	11A/1
SA-522, Type I	K81340	11A/1
SA-553, Type I	K81340	11A/1
SA-553, Type II	K71340	11A/1
SA-645 Grade A	K41583	11A/2

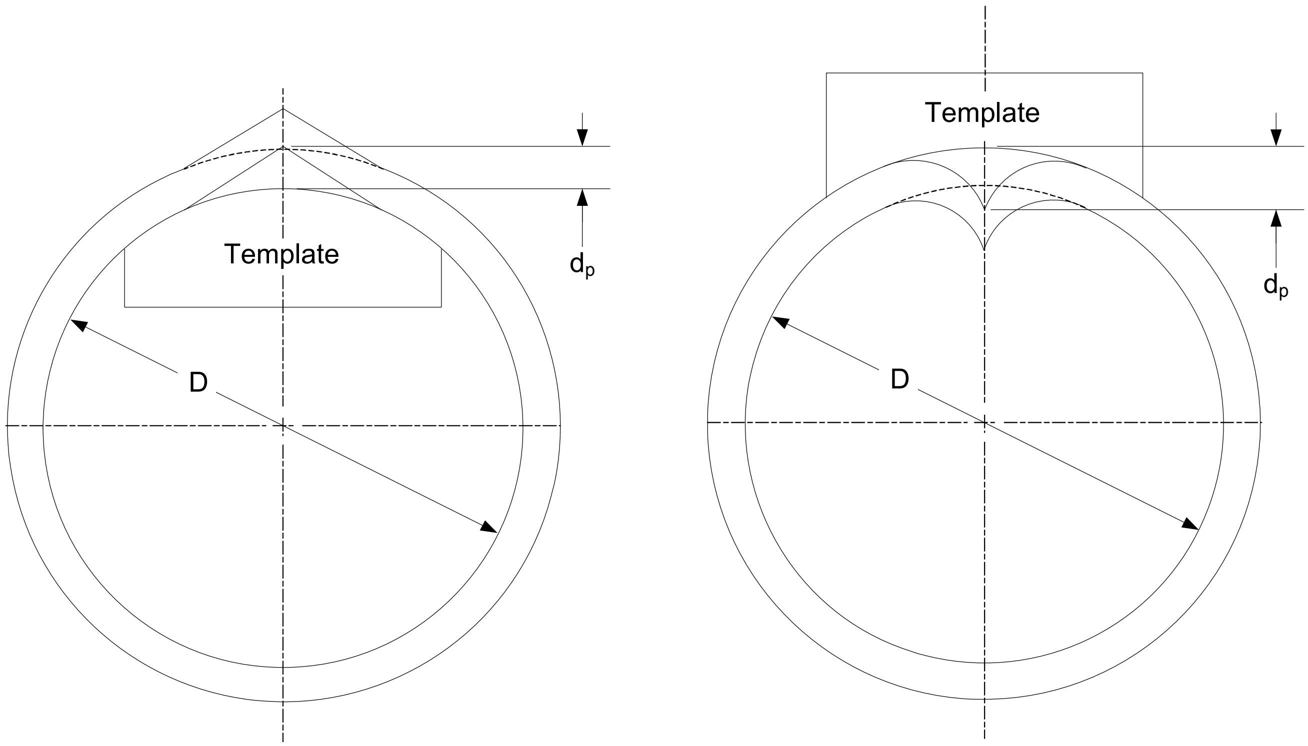
Table 6.19 – High Nickel Alloy Filler For Quench And Tempered Steels

Specification	Classification	F-Number
SFA-5.11	ENiCrFe-2	43
SFA-5.11	ENiCrFe-3	43
SFA-5.11	ENiCrMo-3	43
SFA-5.11	ENiCrMo-6	43
SFA-5.14	ERNiCr-3	43
SFA-5.14	ERNiCrFe-6	43
SFA-5.14	ERNiCrMo-3	43
SFA-5.14	ERNiCrMo-4	43

Table 6.20 – Mandrel Radius for Guided Bend Tests for Forged Fabrication

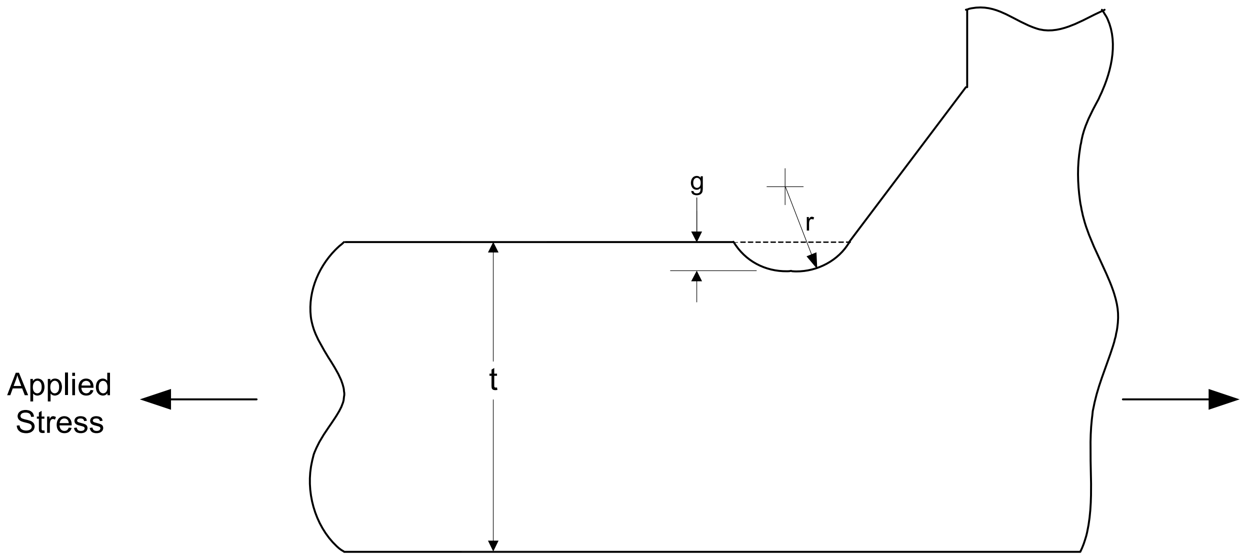
Specimen Thickness	Radius Of Mandrel, B (Note 1)	Radius Of Die, D (Note 1)
10 mm (3/8 in.)	32 mm (1 1/4in.)	37 mm (1 11/16 in.)
t	$\frac{10t}{3}$	$\frac{13t}{3}+2\ mm \quad \left(\frac{13t}{3}+1/8\ in\right)$
Note: The dimension corresponds to dimensions B and D for P-No. 11 material in QW-466.1 of Section IX and other dimensions to be in proportion.		

6.11 Figures



(a) (b)

Figure 6.1 – Peaking Height at a Category A Joint



g = 0.5 mm (0.02 in.) below undercut;
$r \geq 0.25t \geq 4g$

Figure 6.2 – Weld Toe Dressing

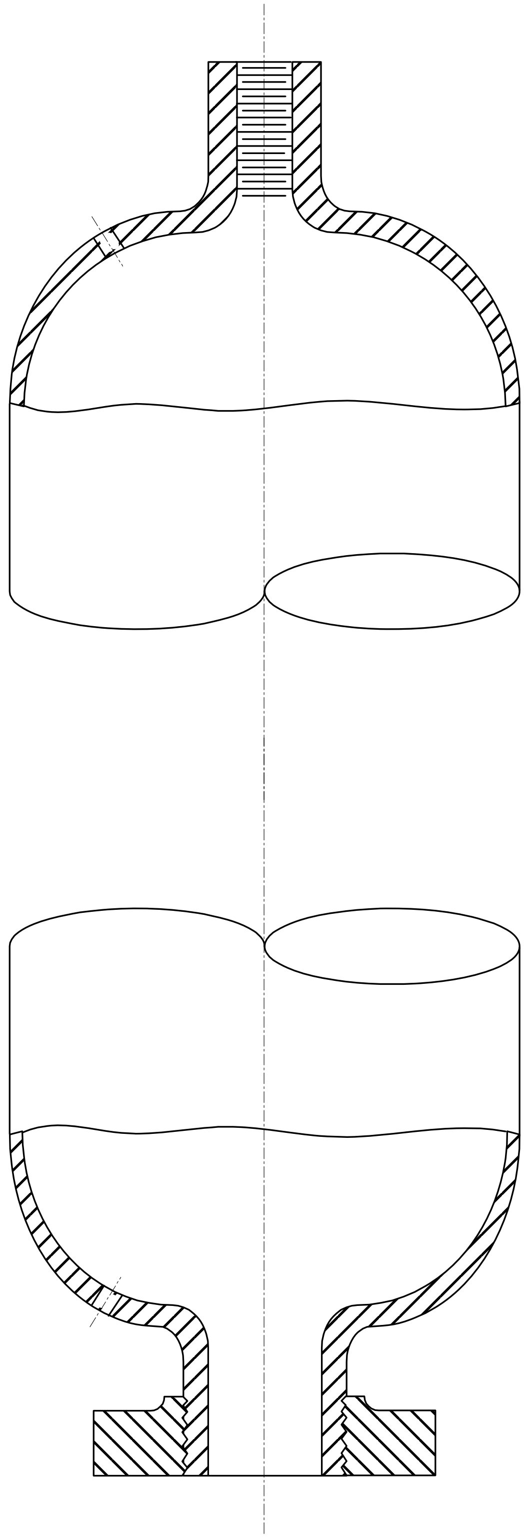
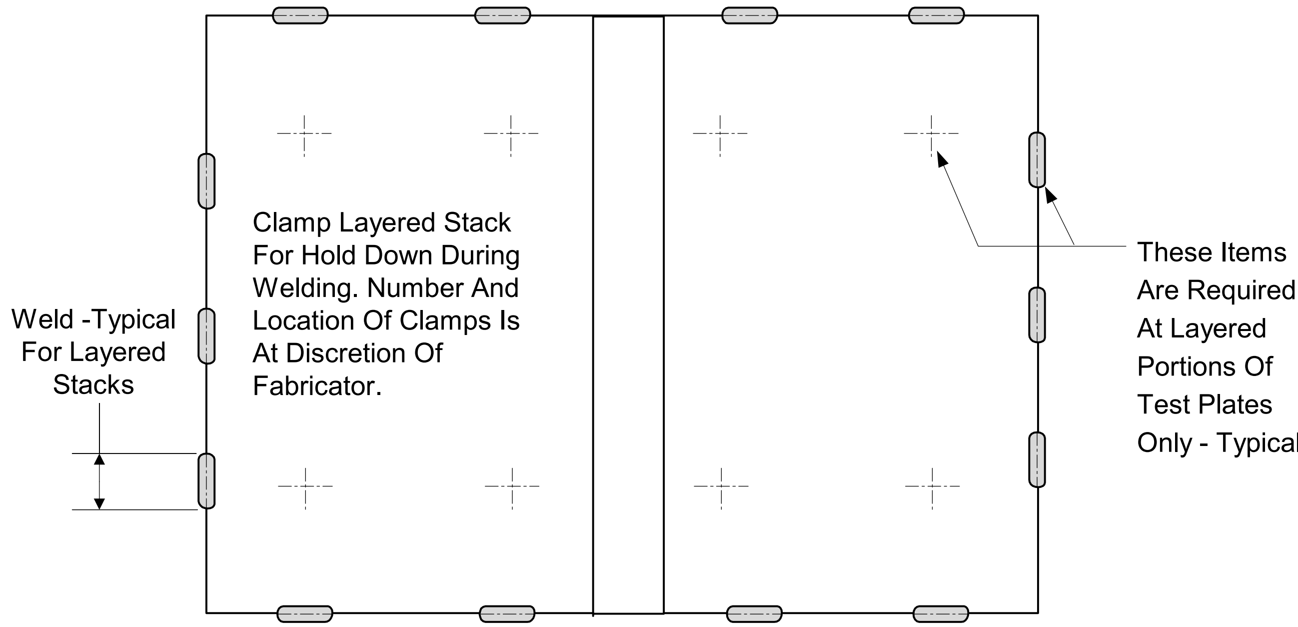
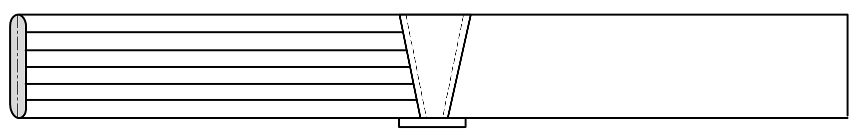


Figure 6.3 – Forged Bottle Construction

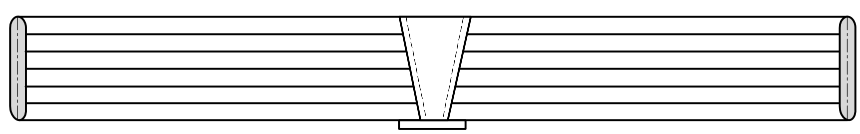


After The Specimen Location Is Laid Out, The Outer Edges Of Layered Stack Shall Be Welded Together In This Location In Order To Prevent Layers From Separating.

Plan View Of Solid To Layered And Layered To Layered Test Plates



Layered To Solid Test Plate



Layered To Layered Test Plate

Figure 6.4 – Solid-To-Layer and Layer-To-Layer Test Plates

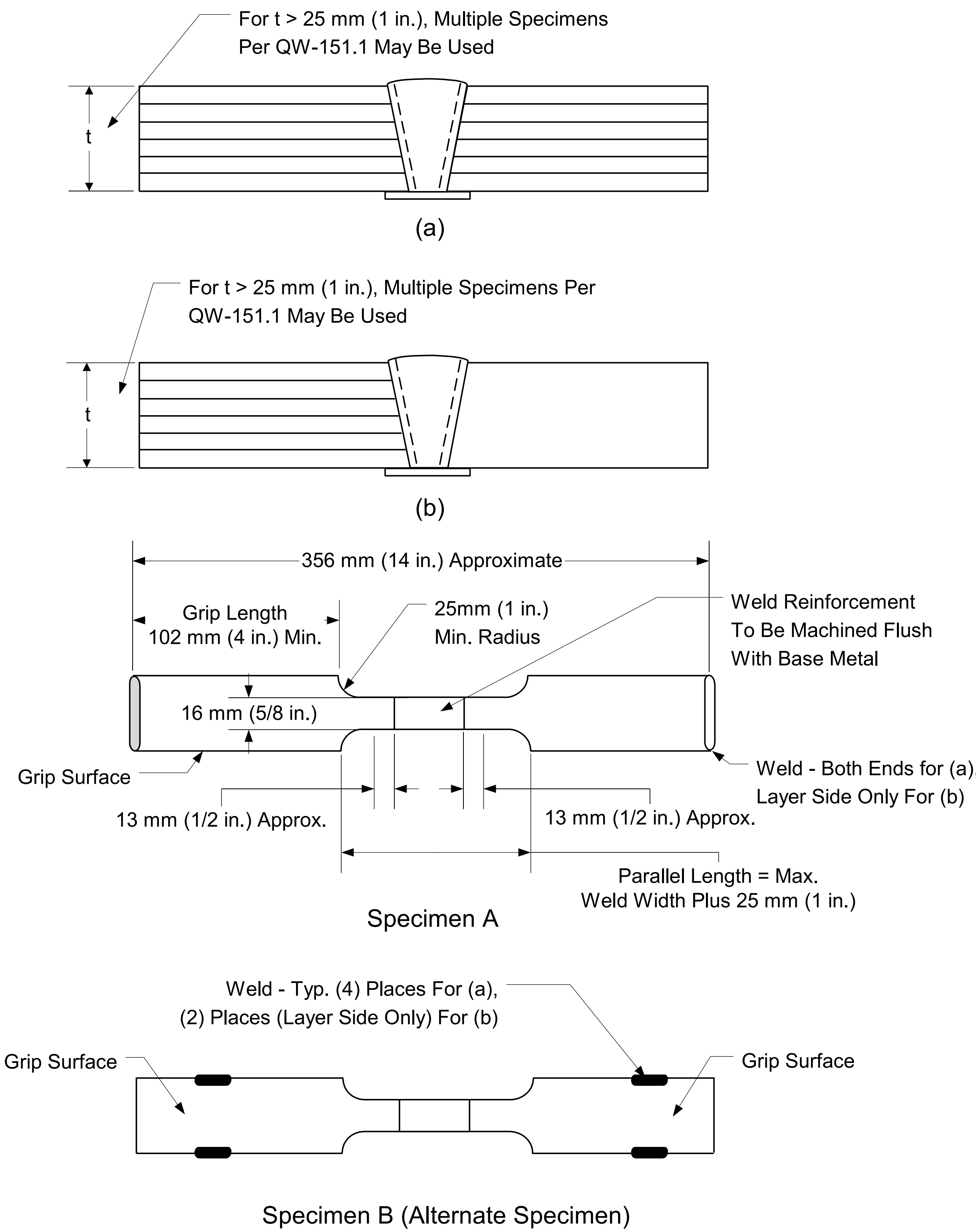


Figure 6.5 – Tensile Specimens for Layered Vessel Construction

PART 7

INSPECTION AND EXAMINATION REQUIREMENTS

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7.1 General

The requirements for examination including Nondestructive Examination (NDE) during fabrication of pressure vessels of welded construction to be marked with Code symbol are provided in this Part. The requirements for examination of materials are provided in Part 3.

7.2 Responsibilities and Duties

7.2.1 Responsibilities and Duties of the Manufacturer and Inspector

Responsibilities and duties of the Manufacturer and Inspector are covered in Annex 7.A.

7.2.2 Access for Inspector

The Manufacturer of the vessel shall arrange for the Inspector to have free access to such parts of all plants as are concerned with the supply or manufacture of materials for the vessel, when so requested. The Inspector shall be permitted free access, at all times while work on the vessel is being performed, to all parts of the Manufacturer's shop that concern the construction of the vessel and to the site of field erected vessels during the period of assembly and testing of the vessel.

7.2.3 Notification of Work Progress

The Manufacturer shall notify the Inspector of the progress of all work associated with the design, fabrication, inspection and examination, and testing of the pressure vessel. In addition, the Manufacturer shall notify the Inspector reasonably in advance when any required tests or inspections are to be performed.

7.3 Verification and Examination Prior To Welding

7.3.1 Compliance of Material with Requirements and Marking

7.3.1.1 All materials used in the construction of the pressure vessel shall comply in all respects with the requirements of this Division.

7.3.1.2 The materials used in the construction of the pressure vessel shall bear the identification required by the applicable material specification, except as otherwise provided in Part 3. Should the identifying marks be obliterated or the material be divided into two or more parts, the marks shall be properly transferred by the Manufacturer as provided in paragraph 3.2.7.2.

7.3.2 Check of Component Parts

Component parts of the pressure vessel shall be checked to verify the following:

a) Head and shell sections conform to the prescribed shape and meet the thickness requirements after forming.

b) Nozzles, manhole frames, lugs, stiffening rings, reinforcement around openings, and other attachments to be welded to the inside or outside surfaces of the vessel properly conform to the contour of the vessel surface.

c) All dimensional checks required by this Division have been performed~~.~~

7.3.3 Verification of Heat Treatment Practice

It shall be verified that all heat treatment operations required by this Division for all materials used in the construction of the pressure vessel are properly conducted.

7.3.4 Verification of Welding Procedure Specification

It shall be verified that the welding procedures employed in construction have been qualified under the provisions of Section IX and as specified in this Division.

7.3.5 Verification of Welder and Welding Operator Performance Qualification

It shall be verified that all welders or welding operators employed for the welding of the pressure vessel have been qualified under the provisions of Section IX.

7.3.6 Qualification of Nondestructive Examination Personnel

a) The Manufacturer shall be responsible for assuring that nondestructive examination (NDE) personnel have been qualified and certified in accordance with their employer’s written practice prior to performing or evaluating examinations required by this Division. SNT-TC-1A or CP-189 shall be used as a guideline for employers to establish their written practice. National or international Central Certification Programs, such as the ASNT Central Certification Program (ACCP), may be used to fulfill the examination and demonstration requirements of the employer’s written practice. Provisions for training, experience, qualification, and certification of NDE personnel shall be described in the Manufacturer’s Quality Control System.

b) NDE personnel shall be qualified by examination. Qualification of NDE Level III personnel certified prior to the 2004 Edition of this Division may be based on demonstrated ability, achievement, education, and experience. Such qualification shall be specifically addressed in the written practice. When NDE personnel have been certified in accordance with a written practice based on an edition of SNT-TC-1A or CP-189 referenced in Table 1.1, their certification shall be valid until their next scheduled recertification.

c) Recertification shall be in accordance with the employer’s written practice based on the edition of SNT-TC-1A or CP-189 referenced in Table 1.1. Recertification may be based on evidence of continued satisfactory performance or by reexamination(s) deemed necessary by the employer.

7.4 Examination of Welded Joints

7.4.1 Nondestructive Examination Requirements

7.4.1.1 All finished welds shall be subject to visual examination in accordance with paragraph 7.5.2.

7.4.1.2 All finished welds shall be subject to nondestructive examination depending on Examination Group selected in paragraph 7.4.2 and the Joint Category and Weld Type as defined in paragraph 4.2.

7.4.1.3 All welding shall be subject to in-process examination by visual examination at the fit-up stage and during back gouging.

7.4.2 Examination Groups for Pressure Vessels

7.4.2.1 Definition of Examination Groups

a) Table 7.1 defines the Examination Groups assigned to welded joints based on the manufacturing complexity of the material group, the maximum thickness, the welding process, and the selected joint efficiency. Nondestructive examination of welded joints shall be performed as indicated for each Examination Group. Each of the Examination Groups are further subdivided into sub-groups “a” and “b” to reflect the crack sensitivity of the material.

b) Table 7.2 indicates the required NDE, joint category designation, joint efficiency, and acceptable joint types for each Examination Group.

c) The *governing welded joint* as used in paragraph 7.4.2.2 below is that welded joint within a given vessel section (e.g., shell course or head) that, as a result of the selected joint efficiency, determines the thickness of that vessel section.

7.4.2.2 Multiple Examination Groups

a) When more than one governing welded joint is located in a pressure vessel, a combination of Examination Groups is permitted, provided that the requirements of Table 7.1 are satisfied.

b) If a combination of Examination Groups are used in a single vessel, the following requirements shall be met:

1) In each vessel section, the Examination Group of the governing welded joint shall be applied to all welds within that vessel section, including nozzle attachment welds.

2) Welds connecting two welded sections that are assigned to different Examination Groups shall be assigned to the Examination Group with the greater extent of examination.

3) Welds connecting a welded section to a seamless section, or welds connecting two seamless sections, shall be assigned to an Examination Group based upon available thickness (thickness at the weld less tolerances and corrosion allowance). When the available thickness is greater than 1.18 (derived from 1/0.85) times the minimum required thickness, Examination Group 3 may be assigned. When the available thickness is less than 1.18 times the minimum required thickness, the Examination Group shall be assigned per Table 7.1.

7.4.3 Extent of Nondestructive Examination

7.4.3.1 The extent of examination given in Table 7.2 is a percentage of the total length of the welded joints under consideration.

7.4.3.2 The examination requirements in Table 7.2 pertain to all butt welded joints.

7.4.3.3 The following welding processes shall be examined over their entire length per the requirements of Table 7.2 and if a radiographic examination was performed, the following welds shall also be ultrasonically examined per paragraph 7.5.4 or 7.5.5:

a) Welds made by the electron beam welding process, and

b) Welds made by the continuous drive friction welding process.

7.4.3.4 The following welding processes shall be radiographically examined per paragraph 7.5.3 and ultrasonically examined per paragraph 7.5.4 or 7.5.5 over their entire length per the requirements of Table 7.2. The ultrasonic examinations shall be done following the grain refining (austenitizing) heat treatment or PWHT:

a) Welds made by the electroslag welding process, and

b) Welds made by the electrogas welding process with any single pass thickness greater than 38 mm (1-1/2 in.) in ferritic materials.

7.4.3.5 If the required extent of examination is less than 100%, the extent and location of nondestructive examination shall be determined by the criteria shown below.

a) For shells, formed heads, communicating chambers and jackets the following requirements apply.

1) Nondestructive examination shall be performed at all intersections of longitudinal and circumferential butt joints. A minimum length of 150 mm (6 in) of the longitudinal seam(s) at these intersections shall be examined. Where the inclusion of all intersections exceeds the percentage in Table 7.2 then this higher value shall apply.

2) If additional examination is required to obtain the percentages required in Table 7.2, additional locations on the butt welded joint selected by the Inspector shall be subject to nondestructive examination.

3) A sufficient number of examinations shall be taken to examine the welding of each welder or welding operator. Under conditions where two or more welders or welding operators make weld

layers in a joint, or on the two sides of a double-welded butt joint, one spot may represent the work of all welders or welding operators who performed welding at the location of the spot.

4) When openings are placed within main welds (longitudinal or circumferential) or within a distance of 13 mm (1/2 in) from the main welds, then the main weld shall be examined for a length of not less than the diameter of the opening on each side of the edge of the openings. These welds shall be included as an addition to the percentage in Table 7.2.

b) Nozzles And Branches Attached To The Vessels – To determine the extent of nondestructive examination, the completed circumferential and longitudinal butt joints of at least one nozzle or branch in each group or partial group shall be examined as shown below.

1) If the extent of examination is 100%, each individual nozzle and branch shall be examined.

2) If the extent of examination is 25%, then one complete nozzle or branch for each group of 4 shall be examined.

3) If the extent of examination is 10%, then one complete nozzle or branch for each group of 10 shall be examined.

c) If the inclusion of the number of complete circumferential and longitudinal butt welds or nozzles exceeds the percentage in Table 7.2, then the higher value shall apply.

7.4.4 Selection of Examination Method for Internal (Volumetric) Flaws

The selection of the examination method for internal flaws (radiographic or ultrasonic) shall be in accordance with Table 7.3. The basis of the selection is the most suitable method to the relevant application in relation to the material type and thickness, or as specified by the owner/user.

NOTE: Considerations such as joint geometry or sensitivity of the material to cracking in the welding process may have an overriding influence, indicating that a method different from that in Table 7.3 should be used. In exceptional cases or where the design or load bearing properties of the joint are critical (particularly for partial penetration joints), it may be necessary to employ both methods from Table 7.3 on the same joint or weld.

7.4.5 Selection of Examination Method for Surface Flaws

For nonmagnetic or partially-magnetic materials, or magnetic materials welded with non-magnetic or partially-magnetic filler metals, Liquid Penetrant Examination in accordance with paragraph 7.5.7 shall be used. For magnetic steels, Magnetic Particle Examination or Liquid Penetrant Examination, in accordance with paragraphs 7.5.6 and 7.5.7 respectively, shall be used as applicable.

7.4.6 Surface Condition and Preparation

The examination surface shall be prepared as necessary so that no surface irregularities or foreign matter interfere with the performance or interpretation of the applicable NDE method.

7.4.7 Supplemental Examination for Cyclic Service

Category A and B welds in vessels for which fatigue analysis is mandatory per paragraph 5.5.2 shall be subject to 100% examination in accordance with the methods specified in Table 7.3. Category C, D, and E welds shall be examined by the magnetic particle or liquid penetrant methods in accordance with paragraphs 7.5.6 and 7.5.7, respectively.

7.4.8 Examination and Inspection of Vessels with Protective Linings and Cladding

7.4.8.1 Examination of Chromium Alloy Cladding or Lining

The joints between chromium alloy cladding layers or liner sheets shall be examined for cracks as follows.

a) Joints welded with straight chromium alloy filler metal shall be examined throughout their full length. Chromium alloy welds in continuous contact with the welds in the base metal shall be examined by the radiographic or ultrasonic methods. Liner welds that are attached to the base metal, but merely cross the seams in the base metal, may be examined by any method that will disclose surface cracks.

b) Joints welded with austenitic chromium-nickel steel filler metal or non-air-hardening nickel-chromium-iron filler metal shall be given a radiographic or ultrasonic spot examination in accordance with the methods described in paragraphs 7.5.3 and 7.5.4. For lined construction, at least one spot examination shall include a portion of the liner weld that contacts weld metal in the base plate. One spot examination shall be completed for every 15 m (50 ft) of weld.

7.4.8.2 Examination of Vessels and Parts

a) Vessels or parts of vessels constructed of clad or weld overlay base material shall have welded joints examined using the radiographic or ultrasonic methods as required in paragraphs 7.4.1 through 7.4.7.

b) Examination of Base Plates Protected by a Strip Covering – If the base material weld in integral or weld metal overlay clad or lined construction is protected by a covering strip or sheet of corrosion resistant material applied over the weld in the base material to complete the cladding or lining, then any radiographic or ultrasonic examination required by paragraphs 7.4.1 through 7.4.7 may be made on the completed weld in the base plate before the covering is attached.

c) Examination of Base Plates Protected by an Alloy Weld – The radiographic or ultrasonic examination required by paragraphs 7.4.1 through 7.4.7 shall be made after the joint, including the corrosion resistant layer, is complete, except that the radiographic or ultrasonic examination may be made on the weld in the base material before the alloy cover weld is deposited, provided the following requirements are met.

   1) The thickness of the base material at the welded joint is not less than that required by the design calculation.

   2) The corrosion resistant alloy weld deposit is non-air-hardening.

   3) The completed alloy weld deposit is examined by liquid penetrant examination in accordance with paragraph 7.5.7.

7.4.8.3 Inspection and Tests

a) General Requirements - The rules in the following paragraphs apply specifically to the inspection and testing of vessels that have clad or weld overlay corrosion resistant linings, and shall be used in conjunction with the general requirements for inspection and testing in Parts 7 and 8, respectively.

b) Tightness of Applied Lining

   1) A test for pressure tightness of the applied lining that will be appropriate for the intended service is recommended, but the details of the test shall be as specified in the Users' Design Specification. NOTE: The test should be such as to assure freedom from damage to the load-carrying base material.

   2) Inspection of Vessel Interior after Test for Tightness – Following the hydrostatic pressure test the interior of the vessel shall be visually examined to determine if there is any seepage of the test fluid through the joints in the lining.

   3) Requirements When Seepage Is Detected – In cases where seepage behind the applied liner is detected, the vessel shall be heated slowly for a sufficient time to drive out all test fluid from behind the applied liner without damage to the liner. After the test fluid is driven out, the lining shall be repaired by welding. The Inspector shall determine if repetition of the radiography, the heat treatment, or the hydrostatic test of the vessel after lining repairs is required to determine whether the repair welds may have caused defects that penetrate into the base material.

7.4.9 Examination and Inspection of Tensile Property Enhanced Q and T Vessels

The following paragraphs are applicable only to vessels constructed of ferritic materials, listed in Table 3.A.2, where the yield and ultimate tensile strength have been enhanced by quenching and tempering.

7.4.9.1 Type No.1 Welded Joint

100% examination per Table 7.3 is required. The examination shall be done after all corrosion resistant alloy cover welding has been deposited.

7.4.9.2 Nozzle Attachment Welds

Nozzle attachment welds as provided for in Table 4.2.13 shall be examined using the radiographic or ultrasonic method in accordance with paragraphs 7.5.3 or 7.5.4 or 7.5.5 (see Table 7.2 and Table 7.3). Nozzle attachment welds in shells over 50 mm (2 in) in thickness in accordance with Table 4.2.10 shall be examined by the radiographic or ultrasonic methods in accordance with paragraphs 7.5.3 or 7.5.4, except that for nozzles having an inside diameter of 50 mm (2 in) or less, the radiographic or ultrasonic examination may be omitted. The required radiographic examination shall be made after all corrosion resistant alloy cover weld has been deposited.

7.4.9.3 Weld Examination

a) Except as permitted in paragraph 7.4.9.3.b, all welds, including welds for attaching nonpressure parts to quenched and tempered steel, shall be examined on all exposed surfaces, after pressure tests, by the magnetic particle method in accordance with paragraph 7.5.6. A magnetization method shall be used that will avoid arc strikes. Crack-like flaws are unacceptable and shall be removed or repaired. The vessel shall be retested in accordance with Part 8 following the repair, and the welds re-examined. For nozzle attachments shown in Table 4.2.10, Details 1, 2, and 8, the exposed cross section of the vessel wall at the opening shall be included in the examination.

b) Alternative Use of Liquid Penetrant Method – As an acceptable alternative to magnetic particle examination or when magnetic particle methods are not feasible because of the nonmagnetic character of the weld deposits, a liquid penetrant method shall be used, see paragraph 7.5.7. For vessels constructed of SA-333 Grade 8, SA-334 Grade 8, SA-353, SA-522, SA-553 Types I and II, and SA-645 Grade A material, the surface examination required in Table 7.2 shall be by the liquid penetrant method either before or after the pressure test. Crack-like flaws are unacceptable and shall be removed or repaired. The vessel shall be retested in accordance with Part 8 following the repair, and the repair welds re-examined.

7.4.9.4 Examination of Corrosion Resistant Overlay Weld Metal

Corrosion resistant overlay weld metal shall be examined by a liquid penetrant method in accordance with paragraph 7.5.7. Crack-like flaws are unacceptable and shall be removed or repaired.

7.4.10 Examination and Inspection of Integrally Forged Vessels

The rules of the following paragraphs apply specifically to the nondestructive examination of integrally forged vessels.

7.4.10.1 Ultrasonic Examination

a) If the vessel is constructed of SA-372 Grade J, Class 110 material, the completed vessel after heat treatment shall be examined ultrasonically in accordance with paragraphs 3.3.3 and 3.3.4. The reference specimen shall have the same nominal thickness, composition, and heat treatment as the vessel it represents. The angle beam examination shall be calibrated with a notch of a depth equal to 5% of the nominal section thickness, a length of approximately 25 mm (1 in), and a width not greater than twice its depth.

b) If the vessel is constructed of SA-723 Class 1, Grades 1, 2, and 3, and SA-723 Class 2, Grades 1, 2, and 3 materials, then the completed vessel shall be examined in accordance with paragraph 3.3.4 regardless of thickness.

c) A vessel is unacceptable if examination results show one or more discontinuities that produce indications exceeding in amplitude the indication from the calibrated notch. Round bottom surface indications such as pits, scores, and conditioned areas exceeding the amplitude of the calibrated notch shall be acceptable if the thickness below the indication is not less than the design wall thickness of the vessel, and its sides are faired to a ratio of not less than three to one.

7.4.10.2 Examination of Weld Repairs

a) For weld repairs of material containing 0.35% Carbon or less, all weld repairs shall be examined by radiography, by a magnetic particle method, or by a liquid penetrant method, in accordance with the

requirements of paragraphs 7.5.3, 7.5.6, or 7.5.7. Weld repairs shall be radiographed when the depth of weld repair exceeds 10 mm (3/8 in) or one-half the material thickness whichever is less. The acceptability of the repair welds shall be determined by the acceptance standards set forth in the applicable paragraph.

b) For weld repairs of material containing more than 0.35% Carbon:

   1) The examination of weld repairs in other than quenched and tempered materials shall meet the requirements of paragraph 7.4.10.2.a, except that radiography shall be required when the depth of weld repair exceeds 6 mm (1/4 in) or one-half the material thickness whichever is less.

   2) Examination of weld repairs in material that is to be or has been liquid quenched and tempered shall meet the requirements of paragraph 7.4.10.2.a, except that radiography shall be required regardless of the depth of weld deposit.

7.4.10.3 Inspection of Test Specimens and Witnessing Tests

When test specimens are to be taken under the applicable material specifications, the Inspector at his option may witness the selection, identifying stamping, and testing of these specimens. Tests and retests shall be made in accordance with the requirements of the material specification.

7.4.11 Examination and Inspection of Fabricated Layered Vessels

7.4.11.1 The rules of the following paragraphs apply specifically to the nondestructive examination of pressure vessels and vessel parts that are fabricated using layered construction. The examination requirements for layered vessel construction are shown in Table 7.4. The rules of paragraphs 7.4.1 through 7.4.7 with Examination Group 1 or 2, whichever is applicable, shall apply for the non-layered parts that are integral with the layered vessel.

7.4.11.2 Inner Shells and Inner Heads

Category A and B joints in the inner shells of layered shell sections and in the inner heads of layered heads before application of the layers shall be examined throughout their entire length by the radiographic or ultrasonic method in accordance with paragraphs 7.5.3 or 7.5.5 (see Table 7.3).

7.4.11.3 Layers – Welded Joints

a) Category A joints in layers 3 mm through 8 mm (1/8 in. through 5/16 in) in thickness welded to the previous surface shall be examined for 100% of their length by the magnetic particle method in accordance with paragraph 7.5.6.

b) Category A joints in layers over 8 mm through 16 mm (5/16 in. through 5/8 in) in thickness welded to the previous surface shall be examined for 100% of their length by the magnetic particle method in accordance with paragraph 7.5.6. In addition, these joints shall be examined for 10% of their length at random using the ultrasonic method in accordance with paragraph 7.5.4, except that for the bottom 10% of the weld thickness the distance amplitude correction curve or reference level may be raised by 6 dB. The random spot examination shall be performed as specified in paragraph 7.4.11.10.

c) Category A joints in layers over 16 mm through 22 mm (5/8 in. through 7/8 in) in thickness welded to the previous layer shall be examined for 100% of their length using the ultrasonic method in accordance with paragraph 7.5.4, except that for the bottom 10% of the weld thickness the distance amplitude correction curve or reference level may be raised by 6 dB.

d) Category A joints in layers not welded to the previous surface shall be examined before assembly for 100% of their length by radiographic or ultrasonic method in accordance with paragraph 7.5.3, 7.5.4 or 7.5.5 (see Table 7.3).

e) Welds in spirally wound strip construction with a winding (spiral angle) of 75° or less measured from the vessel axial centerline shall be classified as Category A joints and examined accordingly.

7.4.11.4 Layers – Step Welded Girth Joints

a) Category B joints in layers 3 mm through 8 mm (1/8 in. through 5/8 in) in thickness shall be examined for 10% of their length by the magnetic particle method (direct current only) in accordance with paragraph 7.5.6. The random spot examination shall be performed as specified in paragraph 7.4.11.10.

b) Category B joints in layers over 8 mm through 16 mm (5/16 in. through 5/8 in) in thickness shall be examined for 100% of their length by the magnetic particle method (using direct current only) in accordance with paragraph 7.5.6.

c) Category B joints in layers over 16 mm through 22 mm (5/8 in. through 7/8 in) in thickness shall be examined for 100% of their length by the magnetic particle method (using direct current only) in accordance with paragraph 7.5.6. In addition these joints shall be examined for 10% of their length by the ultrasonic method in accordance with paragraph 7.5.4, except that for the bottom 10% of the weld thickness the distance amplitude correction curve or reference level may be raised by 6 dB. The random spot examination shall be performed as specified in paragraph 7.4.11.10.

d) Category B joints in layers over 22 mm (7/8 in) in thickness shall be examined for 100% of their length by the ultrasonic method in accordance with paragraph 7.5.4, except that for the bottom 10% of the weld thickness, the distance amplitude correction curve or reference level may be raised by 6 dB.

7.4.11.5 Butt Joints

a) Full thickness welding of solid sections to layered sections. Category A, B, and D joints attaching a solid section to a layered section of any of the layered thicknesses given in paragraph 7.4.11.2 shall be examined by the radiographic method for their entire length in accordance with paragraph 7.5.3.

b) It is recognized that layer wash or acceptable gaps (see paragraph 6.8.8.3) may show as indications difficult to distinguish from slag on the radiograph. Layer wash is defined as the indications resulting from slight weld penetration at the layer interfaces. Acceptance shall be based on reference to the weld geometry as shown in Figure 7.1. As an alternative, an angle radiographic technique, as shown in Figure 7.2, may be used to locate individual gaps in order to determine the acceptability of the indication. Category A and B joints attaching a layered section to a layered section need not be radiographed after being fully welded when the Category A hemispherical head and Category B welded joints of the inner shell or inner head made after application of the layers have been radiographed in accordance with paragraph 7.5.3.

c) The inner shell or inner head thicknesses need not be radiographed in thicknesses over 22 mm (0.875 in) if the completed joint is radiographed. Weld joints in the inner shell or inner head welded after application of the layers of the inner shell or inner head weld joints shall be radiographed throughout their entire length and meet the requirements of paragraph 7.4.11.2.

7.4.11.6 Flat Head and Tubesheet Weld Joints

Category C joints attaching layered shell, or layered heads to flat heads and tubesheets as shown in Figure 4.13.6, shall be examined to the same requirements as specified in paragraph 7.4.11.4 for Category B joints.

7.4.11.7 Nozzle and Communicating Chambers Weld Joints

Category D weld joints in layered shells or layered heads that do not require radiographic examination shall be examined by the magnetic particle or liquid penetrant method in accordance with paragraph 7.5.6 or 7.5.7. The partial penetration weld joining liner type nozzle to layer vessel shells or layer heads, as shown in Figure 4.13.9, shall be examined by magnetic particle or liquid penetrant in accordance with paragraph 7.5.6 or 7.5.7.

7.4.11.8 Welds Attaching Nonpressure Parts and Stiffeners

a) All welds attaching supports, lugs, brackets, stiffeners, and other nonpressure attachments to pressure parts shall be examined on all exposed surfaces by the magnetic particle or liquid penetrant method in accordance with paragraph 7.5.6 or 7.5.7.

b) The examination required in paragraph 6.2.4.7 shall be made after postweld heat treatment for nonpressure parts and stiffeners attached to Material Type 2 parts, see Table 4.2.3.

7.4.11.9 Transition Welds

a) All weld metal buildup in solid wall sections or fillet welds in layered transitions shall be examined over the full surface of the deposit by either the magnetic particle or liquid penetrant method in accordance with paragraph 7.5.6 or 7.5.7.

b) When such surface weld metal buildup is used in welded joints that require radiographic examination, the weld metal buildup shall be included in the examination.

7.4.11.10 Random Spot Examination and Repair of Welds

The random ultrasonic examination of paragraph 7.4.11.3.b and paragraph 7.4.11.4.c, and random magnetic particle examination of paragraph 7.4.11.4.a shall be performed as follows.

a) The location of the random spot shall be chosen by the Inspector, except that when the Inspector has been duly notified in advance and cannot be present or otherwise make the selection, the fabricator may exercise his own judgment in selecting the random spot or spots. The minimum length of a spot shall be 150 mm (6 in).

b) When any random spot examination discloses welding which does not comply with the minimum quality requirements of paragraph 7.4.11.3.b and paragraphs 7.4.11.4.a and 7.4.11.4.c, two additional spots of equal length shall be examined in the same weld unit at locations away from the original spot. The locations of these additional spots shall be determined by the Inspector or fabricator as provided for in the original spot examination.

c) If either of the two additional spots examined shows welding that does not comply with the minimum quality requirements of paragraph 7.4.11.3.b and paragraphs 7.4.11.4.a and 7.4.11.4.c the entire unit of weld represented shall be rejected. The entire rejected weld shall be removed and the joint shall be re-welded or, at the fabricator's option, the entire unit of weld represented shall be completely examined and defective welding only need be corrected.

d) Repair welding shall be performed using a qualified procedure and in a manner acceptable to the Inspector. The re-welded joint or the weld repaired areas shall be random spot examined at one location in accordance with the requirements of paragraph 7.4.11.3.b and paragraphs 7.4.11.4.a and 7.4.11.4.c.

7.5 Examination Method and Acceptance Criteria

7.5.1 General

Nondestructive Examination (NDE) techniques used in this Division and their associated acceptance criteria are shown in Table 7.5.

7.5.2 Visual Examination

7.5.2.1 Examination Method

All welds for pressure retaining parts shall be visually examined. Personnel performing visual examinations shall pass the annual eye test in accordance with paragraph 7.A.2.2.

7.5.2.2 Acceptance Criteria

Welds that are observed to have indications exceeding the criteria given in Table 7.6 are unacceptable. Unacceptable indications shall be removed or reduced to an indication of acceptable size. Whenever an indication is removed by chipping or grinding and subsequent repair by welding is not required, the area shall be blended into the surrounding surface so as to avoid notches, crevices, or corners. Where welding is required after removal of indications, the repair shall be done in accordance with paragraph 6.2.7.

7.5.2.3 Examination of Hidden Weld Seams

Weld seams that will be hidden in the final vessel configuration shall be visually examined for workmanship prior to final assembly, see paragraph 8.2.5.a.3.

7.5.3 Radiographic Examination

7.5.3.1 Examination Method

All welded joints to be radiographed shall be examined and documented in accordance with Article 2 of Section V except as specified below.

a) A complete set of radiographs and records, as described in T-291 and T-292 of Article 2 of Section V, for each vessel or vessel part shall be retained by the Manufacturer in accordance with paragraph 2.3.5.

b) Personnel performing and evaluating radiographic examinations required by this Division shall be qualified and certified in accordance with paragraph 7.3.6.

c) Evaluation of radiographs shall only be performed by RT Level II or III personnel.

d) Demonstration of density and Image Quality Indicator (IQI) image requirements on production or technique radiographs shall be considered satisfactory evidence of compliance with Article 2 of Section V.

e) Final acceptance of radiographs shall be based on the ability to see the prescribed hole (IQI) image and the specified hole or the designated wire of a wire IQI.

7.5.3.2 Acceptance Criteria

Indications shown on the radiographs of welds and characterized as defects are unacceptable under the conditions listed in this paragraph and shall be repaired as provided in paragraph 6.2.7. Repaired welds shall be re-examined, either by radiography in accordance with this paragraph or, at the option of the Manufacturer, ultrasonically in accordance with paragraph 7.5.4 or 7.5.5 and the standards specified in this paragraph. Should ultrasonic examination be performed, this examination method shall be noted under remarks on the Manufacturer's Data Report Form.

a) Linear Indications

   1) Terminology

      Thickness t – the thickness of the weld excluding any allowable reinforcement. For a butt weld joining two members having different thicknesses at the weld, t is the thinner of these two thicknesses. If a full penetration weld includes a fillet weld, the thickness of the fillet throat shall be included in the calculation of t.

   2) Acceptance/Rejection Criteria

      i) Any crack or zone of incomplete fusion or lack of penetration

      ii) Any other linear indication that has a length greater than:

         1. 6 mm (1/4 in) for t less than or equal to 19 mm (3/4 in),
         2. $t/3$ for t greater than 19 mm (3/4 in) and less than or equal to 57 mm (2 1/4 in),
         3. 19 mm (3/4 in) for t greater than 57 mm (2 1/4 in).

iii) Any group of indications in line that has an aggregate length greater than t in a length of $12t$, except when the distance between the successive imperfections exceeds $6L$, where L is the length of the longest imperfection in the group;

iv) Internal root weld conditions are acceptable when the density or image brightness change as indicated in the radiograph is not abrupt. Linear indications on the radiograph at either edge of such conditions shall be evaluated in accordance with the other sections of this paragraph.

b) Rounded Indications

1) Terminology

i) Rounded Indications – indications with a maximum length of three times the width or less on the radiograph are defined as rounded indications. These indications may be circular, elliptical, conical, or irregular in shape and may have tails. When evaluating the size of an indication, the tail shall be included.

ii) Aligned Indications – a sequence of four or more rounded indications shall be considered to be aligned when they touch a line parallel to the length of the weld drawn through the center of the two outer rounded indications.

iii) Thickness t – the thickness of the weld, excluding any allowable reinforcement. For a butt weld joining two members having different thicknesses at the weld, t is the thinner of these two thicknesses. If a full penetration weld includes a fillet weld, the thickness of the fillet throat shall be included in the calculation of t.

2) Acceptance Criteria

i) Rounded Indication Charts – relevant rounded indications characterized as imperfections shall not exceed those shown in Figures 7.5 through 7.10, which illustrate various types of assorted, randomly dispersed and clustered rounded indications for different weld thicknesses greater than 3 mm (1/8 in). The charts for each thickness range represent full-scale 150 mm (6 in) radiographs, and shall not be enlarged or reduced. The distributions shown are not necessarily the patterns that may appear on the radiograph, but are typical of the concentration and size of indications permitted.

ii) Relevant Indications (see Table 7.7 for examples) – only those rounded indications that exceed the following dimensions shall be considered relevant and compared to the acceptance charts for disposition.

1. $t/10$ for t less than 3 mm (1/8 in)
2. 0.4 mm (1/64 in) for t greater than or equal to 3 mm (1/8 in) and less than or equal to 6 mm (1/4 in)
3. 0.8 mm (1/32 in) for t greater than 6 mm (1/4 in) and less than or equal to 50 mm (2 in)
4. 1.5 mm (1/16 in) for t greater than 50 mm (2 in)
5. Maximum Size of Rounded Indication – the maximum permissible size of any indication shall be $t/4$ or 4 mm (5/32 in), whichever is smaller; except that an isolated indication separated from an adjacent indication by 25 mm (1 in) or more may be $t/3$, or 6 mm (1/4 in), whichever is less. For t greater than 50 mm (2 in) the maximum permissible size of an isolated indication shall be increased to 10 mm (3/8 in).
6. Aligned Rounded Indications – aligned rounded indications are acceptable when the summation of the diameters of the indications is less than t in a length of $12t$ (see Figure 7.3). The length of groups of aligned rounded indications and the spacing between the groups shall meet the requirements of Figure 7.4.
7. Clustered Indications – the illustrations for clustered indications show up to four times as many indications in a local area, as that shown in the illustrations for random indications. The length of an acceptable cluster shall not exceed the lesser of 25 mm (1 in) or $2t$.

Where more than one cluster is present, the sum of the lengths of the clusters shall not exceed 25 mm (1 in) in a 150 mm (6 in) length weld.

8. Weld Thickness t less than 3 mm (1/8 in) – for t less than 3 mm (1/8 in) the maximum number of rounded indications shall not exceed 12 in a 150 mm (6 in) length of weld. A proportionally fewer number of indications shall be permitted in welds less than 150 mm (6 in) in length.

iii) Image Density – density or image brightness within the image of the indication may vary and is not a criterion for acceptance or rejection.

iv) Spacing – the distance between adjacent rounded indications is not a factor in determining acceptance or rejection, except as required for isolated indications or groups of aligned indications.

7.5.4 Ultrasonic Examination

7.5.4.1 All welded joints to be ultrasonically examined shall be examined and documented in accordance with Article 4 of Section V except as specified below:

a) A complete set of records, as described in T-491 and T-492 of Article 4 of Section V, for each vessel or vessel part shall be retained by the Manufacturer in accordance with paragraph 2.C.3. In addition, a record of repaired areas shall be noted as well as the results of the reexamination of the repaired areas. The Manufacturer shall also maintain a record from uncorrected areas having responses that exceed 50% of the reference level. This record shall locate each area, the response level, the dimensions, the depth below the surface, and the classification.

b) Personnel performing and evaluating ultrasonic examinations required by this Division shall be qualified and certified in accordance with paragraph 7.3.6

c) Flaw evaluations shall only be performed by UT Level II or III personnel.

d) Ultrasonic examination shall be performed in accordance with a written procedure certified by the Manufacturer to be in accordance with the requirements of T-150 of Section V.

7.5.4.2 Acceptance Criteria

These standards shall apply unless other standards are specified for specific applications within this Division. All imperfections that produce an amplitude greater than 20% of the reference level shall be investigated to the extent that the operator can determine the shape, identity, and location of all such imperfections and evaluate them in terms of the acceptance standards given in (a) and (b) below.

a) Imperfections that are interpreted to be cracks, lack of fusion, or incomplete penetration are unacceptable regardless of length.

b) All other linear type imperfections are unacceptable if the amplitude exceeds the reference level and the length of the imperfection exceeds the following:

1) 6 mm (1/4 in) for t less than 19 mm (3/4 in)

2) $t/3$ for t greater than or equal to 19 mm (3/4 in) and less than or equal to 57 mm (2-1/4 in)

3) 19 mm (3/4 in) for t greater than 57 mm (2-1/4 in)

In the above criteria, t is the thickness of the weld, excluding any allowable reinforcement (see paragraph 6.2.4.1.d). For a butt weld joining two members having different thicknesses at the weld, t is the thinner of these two thicknesses. If a full penetration weld includes a fillet weld, the thickness of the throat of the fillet shall be included in t.

7.5.5 Ultrasonic Examination Used in Lieu of Radiographic Examination

7.5.5.1 When used in lieu of the radiographic examination requirements of paragraph 7.5.3, ultrasonic examination shall be performed in accordance with a written procedure conforming to the requirements of Section V, Article 4, Mandatory Appendix VIII and the following additional requirements. For SAW welds in 2¼Cr-1Mo-V vessels, ultrasonic examination in accordance with this paragraph are required.

NOTE: For SAW welds in 2¼Cr-1Mo-V vessels, the potential exists for fabrication related cracking. Careful selection of examination techniques, scans, calibrations, and acceptance criteria are necessary to provide the sensitivity required to detect this cracking.

a) The ultrasonic examination area shall include the volume of the weld, plus 50 mm (2 in) on each side of the weld for material thickness greater than 200 mm (8 in). For material thickness 200 mm (8 in) or less, the ultrasonic examination area shall include the volume of the weld, plus the lesser of 25 mm (1 in) or t on each side of the weld. Alternatively, examination volume may be reduced to include the actual heat affected zone (HAZ) plus 6 mm (1/4 in) of base material beyond the heat affected zone on each side of the weld provided the following requirements are met:

   1) The extent of the weld HAZ is measured and documented during the weld qualification process; and

   2) The ultrasonic transducer positioning and scanning device is controlled using a reference mark (paint or low stress stamp adjacent to the weld) to ensure that the actual HAZ plus an additional 6 mm (0.25 in) of base metal is examined.

b) The initial straight beam material examination (T-472 of Section V, Article 4) for reflectors that could interfere with the angle beam examination shall be performed:

   1) Manually,

   2) As part of a previous manufacturing process, or

   3) During the automatic UT examination provided detection of these reflectors is demonstrated.

c) Personnel performing and evaluating UT examinations shall be qualified and certified in accordance with paragraph 7.3.6. Only UT Level II or III personnel shall analyze the data or interpret the results.

d) Contractor qualification records of certified personnel shall be approved by the Certificate Holder and maintained by their employer.

e) In addition, personnel who acquire and analyze UT data shall participate in the qualification of the procedure per Section V, Article 4 Mandatory Appendix IX.

f) The final data package shall be reviewed by a UT Level III individual. The review shall include:

   1) The ultrasonic data record

   2) Data interpretations

   3) Flaw evaluations/characterizations performed by another qualified Level II or III individual. The data review may be performed by another individual from the same organization. Alternatively, the review may be achieved by arranging for a data acquisition and initial interpretation by a Level II individual qualified in accordance with paragraphs 7.3.6 and Section V, Article 4 Mandatory Appendix IX, and a final interpretation and evaluation shall be performed by a Level III individual qualified similarly. The Level III individual shall have been qualified in accordance with paragraph 7.3.6 including a practical examination on flawed specimens.

g) Application of automated ultrasonic examinations shall be noted on the Manufacturer's Data Report, as well as the extent of its use.

NOTE: Sectional scans (S-scans) with phased arrays may be used for the examination of welds, provided they are qualified satisfactorily in accordance with paragraph 7.5.5.1.e. S-scans provide a fan beam from a single emission point, which covers part or all of the weld, depending on transducer size, joint geometry, and section thickness. While S-scans can demonstrate good detectability from side drilled holes, because they

are omni-directional reflectors, the beams can be mis-oriented for planar reflectors (e.g., lack of fusion and cracks.) This is particularly true for thicker sections, and it is recommended that multiple linear passes with S-scans be utilized for components greater than 25mm (1 in.) thick. An adequate number of flaws should be used in the demonstration block to ensure detectability for the entire weld volume.

7.5.5.2 Flaw Sizing

The dimensions of the flaw shall be determined by the rectangle that fully contains the area of the flaw (see Figures 7.11 through 7.15).

a) The length (l) of the flaw shall be drawn parallel to the inside pressure-retaining surface of the component.

b) The depth of the flaw shall be drawn normal to the inside pressure retaining surface and shall be denoted as "a" for a surface flaw or "2a" for a subsurface flaw.

7.5.5.3 Flaw Evaluation and Acceptance Criteria

Flaws shall be evaluated for acceptance using the applicable criteria of Tables 7.8, 7.9 or 7.10, and with the following additional requirements. Unacceptable flaws shall be repaired and the repaired welds shall be re-evaluated for acceptance.

a) Surface Flaws – Flaws identified as surface flaws during the UT examination may or may not be surface connected. Therefore, unless the UT data analysis confirms that the flaw is not surface connected, it shall be considered surface connected or a flaw open to the surface, and is unacceptable unless surface examination is performed. If the flaw is surface connected, the requirements above still apply. However, in no case shall the flaw exceed the acceptance criteria in this Division for the material employed. Acceptance surface examination techniques are as follows:

   1) Magnetic particle examination (MT) in accordance with paragraph 7.5.6,

   2) Liquid penetrant examination (PT) in accordance with paragraph 7.5.7,

   3) Eddy Current examination (ET) in accordance with paragraph 7.5.8.

b) Multiple Flaws

   1) Discontinuous flaws shall be considered a singular planar flaw if the distance between adjacent flaws is equal to or less than the dimension S as shown in Figure 7.12.

   2) Discontinuous flaws that are oriented primarily in parallel planes shall be considered a singular planar flaw if the distance between the adjacent planes is equal to or less than 13 mm (1/2 in) (see Figure 7.13).

   3) Discontinuous flaws that are coplanar and nonaligned in the through-wall thickness direction of the component shall be considered a singular planar flaw if the distance between adjacent flaws is equal to or less than S as shown in Figure 7.14.

   4) Discontinuous flaws that are coplanar in the through-wall direction within two parallel planes 13 mm (1/2 in) apart (i.e., normal to the pressure-retaining surface of the component) are unacceptable if the additive flaw depth dimension of the flaws exceeds those shown in Figure 7.15.

c) Subsurface Flaws – the flaw length (l) shall not exceed 4t.

7.5.6 Magnetic Particle Examination (MT)

7.5.6.1 All magnetic particle examinations shall be performed and documented in accordance with Article 7 of ASME Section V except as specified below:

a) A complete set of records, as described in T-791 and T-792 of Article 7 of Section V, for each vessel or vessel part shall be retained by the Manufacturer until the Manufacturer's Data Report has been signed by the Inspector.

b) Personnel performing and evaluating magnetic particle examinations required by this Division shall be qualified and certified in accordance with paragraph 7.3.6. Evaluation of magnetic particle examination shall only be performed by MT Level II or III personnel.

c) Magnetic particle examination shall be performed in accordance with a written procedure, certified by the Manufacturer to be in accordance with the requirements of T-150 of Section V.

d) Indications will be revealed by retention of magnetic particles. All such indications are not necessarily imperfections, however, since excessive surface roughness, magnetic permeability variations (such as the edge of heat affected zones), etc., may produce similar indications. An indication of an imperfection may be larger than the imperfection that causes it; however, the size of the indication is the basis for acceptance evaluation. Only indications which have any dimension greater than 1.5 mm (1/16 in) shall be considered relevant.

   1) A linear indication is one having a length greater than three times the width.

   2) A rounded indication is one of circular or elliptical shape with a length equal to or less than three times its width.

   3) Any questionable or doubtful indications shall be reexamined to determine whether or not they are relevant.

7.5.6.2 Acceptance Criteria

The following acceptance standards shall apply unless other more restrictive standards are specified for specific material or applications within this Division. Unacceptable indications shall be removed or reduced to an indication of acceptable size. Whenever an indication is removed by chipping or grinding and subsequent repair by welding in not required, the excavated area shall be blended into the surrounding surface so as to avoid notches, crevices, or corners. Where welding is required after removal of indications, the repair shall be done in accordance with paragraph 6.2.7.

a) All surfaces to be examined shall be free of:

   1) Relevant linear indications

   2) Relevant rounded indications greater than 5 mm (3/16 in)

   3) Four or more relevant rounded indications in a line separated by 1.5 mm (1/16 in) or less, edge-to-edge

b) Crack like indications detected, irrespective of surface conditions, are unacceptable.

7.5.7 Liquid Penetrant Examination (PT)

7.5.7.1 All liquid penetrant examinations shall be performed and documented in accordance with Article 6 of ASME Section V except as specified below:

a) A complete set of records, as described in T-691 and T-692 of Article 6 Section V, for each vessel or vessel part shall be retained by the Manufacturer until the Manufacturer's Data Report has been signed by the Inspector.

b) Personnel performing and evaluating liquid penetrant examinations required by this Division shall be qualified and certified in accordance with paragraph 7.3.6. Evaluation of liquid penetrant examination shall only be performed by PT Level II or III personnel.

c) Liquid penetrant examination shall be performed in accordance with a written procedure, certified by the Manufacturer to be in accordance with the requirements of T-150 of Section V.

d) An indication of an imperfection may be larger than the imperfection that causes it; however, the size of the indication is the basis for acceptance evaluation. Only indications with major dimensions greater than 1.5 mm (1/16 in) shall be considered relevant.

   1) A linear indication is one having a length greater than three times the width.

2) A rounded indication is one of circular or elliptical shape with a length equal to or less than three times its width.

3) Any questionable or doubtful indications shall be reexamined to determine whether or not they are relevant.

7.5.7.2 Acceptance Criteria

The following acceptance standards shall apply unless other more restrictive standards are specified for specific material or applications within this Division. Unacceptable indications shall be removed or reduced to an indication of acceptable size. Whenever an indication is removed by chipping or grinding and subsequent repair by welding is not required, the excavated area shall be blended into the surrounding surface so as to avoid notches, crevices, or corners. Where welding is required after removal of indications, the repair shall be done in accordance with paragraph 6.2.7.

a) All surfaces to be examined shall be free of:

1) Relevant linear indications

2) Relevant rounded indications greater than 5 mm (3/16 in)

3) Four or more relevant rounded indications in a line separated by 1.5 mm (1/16 in) or less, edge-to-edge

b) Crack like indications detected, irrespective of surface conditions, are unacceptable

7.5.8 Eddy Current Surface Examination Procedure Requirements (ET)

7.5.8.1 All eddy current examinations shall be performed and documented as described in this section:

a) A complete set of records for each vessel or vessel part shall be retained by the Manufacturer until the Manufacturer's Data Report has been signed by the Inspector.

b) Personnel performing and evaluating eddy current examinations required by this Division shall be qualified and certified in accordance with paragraph 7.3.6. Evaluation of eddy current examination shall only be performed by ET Level II or III personnel.

c) Eddy current examinations shall be performed in accordance with a written procedure, certified by the Manufacturer to be in accordance with the requirements of T-150 of Section V.

7.5.8.2 Procedure Requirements

The procedure shall provide a statement of scope that specifically defines the limits of procedure applicability (e.g., material specification, grade, type, or class). The procedure shall reference a technique specification, delineating the essential variables, qualified in accordance with the requirements below.

7.5.8.3 Procedure Specifications

a) The eddy current procedure shall specify the following regarding data acquisition:

1) instrument or system, including manufacturer's name and model

2) size and type of probe, including manufacturer's name and part number

3) analog cable type and length

4) examination frequencies, or minimum and maximum range, as applicable

5) coil excitation mode (e.g., absolute or differential)

6) minimum data to be recorded

7) method of data recording

8) minimum digitizing rate (samples per inch) or maximum scanning speed (for analog systems), as applicable

9) scan pattern, when applicable (e.g., helical pitch and direction, rectilinear rotation, length, scan index, or overlap)

10) magnetic bias technique, when applicable

11) material type

12) coating type and thickness, when applicable

b) The eddy current procedure shall define the following regarding data analysis:

1) method of calibration (e.g., phase angle or amplitude adjustments)

2) channel and frequencies used for analysis

3) extent or area of the component evaluated

4) data review requirements (e.g., secondary data review, computer screening)

5) reporting requirements (i.e., signal-to-noise threshold, voltage threshold, flaw depth threshold)

6) methods of identifying flaw indications and distinguishing them from nonrelevant indications, such as indications from probe lift-off or conductivity and permeability changes in weld material

7) manufacturer and model of eddy current data analysis equipment, as applicable

8) manufacturer, title, and version of data analysis software, as applicable

c) The procedure shall address requirements for system calibration. Calibration requirements include those actions required to ensure that the sensitivity and accuracy of the signal amplitude and time outputs of the examination system, whether displayed, recorded, or automatically processed, are repeatable and correct. Any process of calibrating the system is acceptable; a description of the calibration process shall be included in the procedure.

d) Data acquisition and analysis procedures may be combined or separate, provided the above requirements are met.

7.5.8.4 Additional Personnel Requirements

a) Personnel performing data acquisition shall have received specific training and shall be qualified by examination, in accordance with the employer's written practice, in the operation of the equipment, applicable techniques, and recording of examination results.

b) Personnel performing analysis of data shall have received additional specific training in the data analysis techniques used in the procedure qualification and shall successfully complete the procedure qualification described below.

c) American Society of Nondestructive Testing (ASNT) standards SNT-TC-1A or CP 189 shall be used as a guideline.

d) Personnel qualifications may be combined provided all requirements are met.

7.5.8.5 Procedure Qualification

a) Data sets for detection and sizing shall meet requirements shown below.

b) The eddy current procedure and equipment shall be considered qualified upon successful completion of the procedure qualification.

c) Essential Variables – an essential variable is a procedure, software, or hardware item that, if changed, could result in erroneous examination results. Further, any item that could decrease the signal to noise ratio to less than 2:1 shall be considered an essential variable.

d) Any two procedures with the same essential variables are considered equivalent. Equipment with essential variables that vary within the demonstrated ranges identified in the Data Acquisition Procedure Specification shall be considered equivalent. When the procedure allows more than one value or range

for an essential variable, the qualification test shall be repeated at the minimum and maximum value for each essential variable with all other variables remaining at their nominal values. Changing essential variables may be accomplished during successive procedure qualifications involving different personnel; each data analyst need not demonstrate qualification over the entire range of every essential variable.

7.5.8.6 Qualification Requirements

a) Specimens to be used in the qualification test shall meet the requirements listed herein unless a set of test specimens is designed to accommodate specific limitations stated in the scope of the examination procedure (e.g., surface roughness or contour limitations). The same specimens may be used to demonstrate both detection and sizing qualification. For examination of vessels with coated surfaces, Section V, Article 8 shall apply.

b) Specimens shall be fabricated from the same base material nominal composition (UNS Number) and heat treatment (e.g., solution annealed, precipitation hardened, solution heat treated and aged) as those to be examined.

c) Specimen surface roughness and contour shall be generally representative of the surface roughness and contour of the component surface to be examined. The examination surface curvature need not be simulated if the ratio of the component diameter to the coil diameter exceeds 20:1.

d) Welding shall be performed with the same filler material AWS classification, and postweld heat treatment *(e.g.,* as welded, solution annealed, stress relieved) as the welds to be examined.

e) Defect Conditions

   1) The qualification flaws shall be cracks or notches.

   2) The length of cracks or notches open to the surface shall not exceed 3.2 mm (0.125 in.).

   3) The maximum depth of a crack or compressed notch shall be 1.02 mm (0.040 in.).

   4) Machined notches shall have a maximum width of 0.25 mm (0.010 in.) and a maximum depth of 0.51 mm (0.020 in.).

f) Demonstration Specimens – the demonstration specimen shall include one crack or notch at each of the following locations:

   1) on the weld

   2) in the heat affected zone

   3) at the fusion line of the weld

   4) in the base material

g) Procedure Qualification Acceptance Criteria. All flaws in each of the four identified areas shall be detected with a minimum 2:1 signal-to-noise ratio at the maximum digitization rate (for digital systems) or maximum scanning speed (for analog systems) permitted by the procedure.

7.5.8.7 Evaluation of Eddy Current Results

Eddy current results shall be evaluated in accordance with the qualified procedure described in paragraph 7.5.8.3.b. If a flaw is determined by ET to be surface connected it shall comply with the Acceptance Criteria in paragraph 7.5.8.8 below.

7.5.8.8 Acceptance Standards

These acceptance standards apply unless other more restrictive standards are specified for specific materials or applications within this Division. All surfaces examined shall be free of relevant ET surface flaw indications.

7.5.9 Evaluation and Retest for Partial Examination

The locations selected under paragraphs 7.4.3.5.a and 7.4.3.5.b shall be deemed to be representative of the welds examined. An imperfection detected on the circumferential seam shall be considered as representing

the condition of the whole circumferential seam. An imperfection detected on the longitudinal seam shall be considered as representing the condition of the whole longitudinal seam. An imperfection detected on a nozzle or branch shall be considered as representing the condition of the group of nozzles or branches. According to the imperfections type, retesting shall be as follows:

a) When a percentage of the weld, defined in Table 7.2, is examined and meets the minimum quality requirements of paragraph 7.5.3.2, the entire weld length represented by this examination is acceptable.

b) When a radiograph representing a percentage of weld, as defined in Table 7.2 and examined as required in paragraph 7.4.3.5, discloses welding which does not comply with the minimum quality requirements of paragraph 7.5.3.2, two additional welds deposited by the same welder which are of the same type and category and were not previously examined shall be radiographically examined. The additional welds to be examined shall be selected by the Inspector or fabricator under the same criteria applied to the original examination.

    1) If the two additional welds examined are acceptable in accordance with the minimum quality requirements of paragraph 7.5.3.2, the entire weld increment represented by the radiographs is acceptable, provided the unacceptable indications disclosed by radiographs are removed, repaired, and reexamined.

    2) If either of the two additional welds examined do not comply with the minimum quality requirements of paragraph 7.5.3.2, the entire increment of weld represented shall be repaired and reexamined; or, at the fabricator's option, the entire increment of weld represented by the unacceptable radiographs shall be completely radiographed and all unacceptable indications repaired and reexamined.

    3) Repair welding shall be performed using a qualified procedure and deposited by a qualified welder. The rewelded joint, or the weld repaired areas, shall be radiographically examined as provided for in Table 7.2.

7.6 Final Examination of Vessel

7.6.1 Surface Examination after Hydrotest

If a fatigue analysis is required for a part of a vessel, then all of the internal and external surfaces of pressure boundary and attachment welds for that part shall be examined by wet magnetic particle method (if ferromagnetic) or by liquid penetrant method (if nonmagnetic) after hydrotest, unless accessibility prevents meaningful interpretation and characterization of imperfections. The acceptance criteria shall be paragraphs 7.5.6 and 7.5.7.

7.6.2 Inspection of Lined Vessel Interior after Hydrotest

When it is observed that the test fluid seeps behind the applied liner during or after hydrotest, the fluid shall be driven out and the lining shall be repaired by welding in accordance with paragraph 7.4.8.3.b.3.

7.7 Leak Testing

When specified in the Users' Design Specification, leak testing shall be carried out in accordance with Article 10 of Section V in addition to hydrostatic test as per paragraph 8.2 or pneumatic test as per paragraph 8.3.

7.8 Acoustic Emission

If specified in the Users' Design Specification, acoustic emission examination shall be carried out in accordance with Article 12 of Section V during the hydrostatic test or pneumatic test. The acceptance criteria shall be as stated in the Users' Design Specification.

7.9 Tables

Table 7.1 – Examination Groups For Pressure Vessels

Parameter	Examination Group (1)					
	1a	1b	2a	2b	3a	3b
Permitted Material (1) (2)	**All Materials in Annex 3.A**	**P-No.1 Gr 1 and 2, P-No. 8 Gr 1**	**P-No. 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No 11A Gr 1 P-No. 11A Gr 2 P-No. 10H Gr 1**	**P-No.1 Gr 1 and 2, P-No. 8 Gr 1**	**P-No. 8 Gr 2, P-No 9A Gr 1, P-No 9B Gr 1, P-No. 10H Gr 1**	**P-No.1 Gr 1 and 2, P-No. 8 Gr 1**
Maximum thickness of governing welded joints	Unlimited (4)		30mm (1 3/16 in) for P-No 9A Gr 1 and P-No 9B Gr 1; 16mm (5/8 in) for P-No. 8,Gr 2 (5) P-No. 11A Gr 1 P-No. 11A Gr 2 P-No. 10H Gr 1	50mm (2 in) for P-No.1 Gr 1 and P-No. 8 Gr 1; 30mm (1 3/16 in) for P-No.1 Gr 2	30mm (1 3/16 in) for P-No. 9A Gr 1 and P-No. 9B Gr 1; 16mm (5/8 in) for P-No.8, Gr 2 (5) P-No. 10H Gr 1	50mm (2 in) for P-No.1 Gr 1 and P-No. 8 Gr 1; 30mm (1 3/16 in) for P-No.1 Gr 2
Welding process	Unrestricted (4)		Mechanized Welding Only (3)		Unrestricted (4)	
Design Basis (6)	Part 4 or Part 5 of this Division		Part 4 or Part 5 of this Division		Part 4 of this Division	

Notes:

1. All Examination Groups require 100% visual examination to the maximum extent possible.
2. See Part 3 for permitted material.
3. Mechanized means machine and/or automatic welding methods.
4. Unrestricted with respect to weld application modes as set forth in this Table.
5. See Table 7.2 for NDE, joint category, and permissible weld joint details that differ between Examination Groups 1a and 1b.
6. The design basis is the analysis method used to establish the wall thickness.

Table 7.2 – Nondestructive Examination

Examination Group					1a	1b	2a	2b	3a	3b
Permitted Materials					**All Materials in Annex 3.A**	**P-No 1 Gr 1 & 2 P-No. 8 Gr 1**	**P-No. 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No. 11A Gr 1 P-No. 11A Gr 2 P-No. 10H Gr 1**	**P-No 1 Gr 1 & 2 P-No 8 Gr 1**	**P-No 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No 10H Gr 1**	**P-No 1 Gr 1 & 2 P-No 8 Gr 1**
Weld Joint Efficiency					**1.0**	**1.0**	**1.0**	**1.0**	**0.85**	**0.85**
Joint Category	**Type of Weld (1)**			**Type of NDE (2)**	**Extent of NDE (10)(11)(12)**					
A	Full penetration butt weld	1	Longitudinal joints	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
B		1	Circumferential joints on a shell	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	10% 10%	10% (3) 10% (4)
B		2,3	Circumferential joints on a shell with backing strip (9)	RT or UT MT or PT	NA NA	100% 10%	NA NA	25% 10%	NA NA	25% 10%
B		1	Circumferential joints on a nozzle where , d>150mm (6 in) or t>16mm (5/8 in)	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	10% 10%	10% (3) 10% (4)
B		2,3	Circumferential joints on a nozzle where d > 150 mm (6 in) or t > 16 mm (5/8 in) with backing strip (9)	RT or UT MT or PT	NA NA	100% 10%	NA NA	25% 10%	NA NA	25% 10%
B		1	Circumferential joints on a nozzle where d ≤ 150 mm (6 in) and t ≤ 16 mm (5/8 in)	MT or PT	100%	10%	100%	10%	10%	10%
A		1	All welds in spheres, heads and hemispherical heads to shells	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
B		1	Attachment of a conical shell with a cylindrical shell at an angle ≤ 30	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	10% 10%	10% 10% (4)
B		8	Attachment of a conical shell with a cylindrical shell at an angle > 30	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)

Table 7.2 – Nondestructive Examination

Examination Group					1a	1b	2a	2b	3a	3b
Permitted Materials					**All Materials in Annex 3.A**	**P-No 1 Gr 1 & 2 P-No. 8 Gr 1**	**P-No. 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No. 11A Gr 1 P-No. 11A Gr 2 P-No. 10H Gr 1**	**P-No 1 Gr 1 & 2 P-No 8 Gr 1**	**P-No 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No 10H Gr 1**	**P-No 1 Gr 1 & 2 P-No 8 Gr 1**
Weld Joint Efficiency					**1.0**	**1.0**	**1.0**	**1.0**	**0.85**	**0.85**
Joint Category	**Type of Weld (1)**			**Type of NDE (2)**	**Extent of NDE (10)(11)(12)**					
C	Assembly of a flat head or tubesheet, with a cylindrical shell or Assembly of a flange or a collar with a shell	1,2, 3, 7	With full penetration	UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
C		9, 10	With partial penetration if a > 16 mm (5/8 in) (16)	UT MT or PT	NA	NA	NA	NA	25% 10%	10% 10%
C		9, 10	With partial penetration if $a \le$ 16 mm (5/8 in) (16)	UT MT or PT	NA	NA	NA	NA	10%	10%
C	Assembly of a flange or a collar with a nozzle	1,2, 3, 7	With full penetration	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
C		9, 10	With partial penetration	MT or PT	NA	NA	NA	NA	10%	10%
C		9, 10	With full or partial penetration $d \le$ 150 mm (6 in) and $t \le$ 16 mm (5/8 in)	MT or PT	10%	10% (4)	10%	10% (4)	10%	10% (4)
D	Nozzle or branch (5)	1,2, 3, 7	With full penetration d > 150 mm (6 in) or t > 16 mm (5/8 in)	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
D		1,2, 3, 7	With full penetration $d \le$ 150 mm (6 in) and $t \le$ 16 mm (5/8 in)	MT or PT	100%	10%	100%	10%	10%	10%
D		9, 10	With partial penetration for any d a > 16 mm (5/8 in) (17)	UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
D		9, 10	With partial penetration d > 150 mm (6 in) $a \le$ 16 mm (5/8 in) (17)	MT or PT	NA	NA	NA	NA	10%	10%
D		9, 10	With partial penetration $d \le$ 150 mm (6 in) $a \le$ 16 mm (5/8 in)	MT or PT	100%	10%	100%	10%	10%	10%

Table 7.2 – Nondestructive Examination

Examination Group					1a	1b	2a	2b	3a	3b
Permitted Materials					All Materials in Annex 3.A	P-No 1 Gr 1 & 2 P-No. 8 Gr 1	P-No. 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No. 11A Gr 1 P-No. 11A Gr 2 P-No. 10H Gr 1	P-No 1 Gr 1 & 2 P-No 8 Gr 1	P-No 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No 10H Gr 1	P-No 1 Gr 1 & 2 P-No 8 Gr 1
Weld Joint Efficiency					1.0	1.0	1.0	1.0	0.85	0.85
Joint Category	Type of Weld (1)			Type of NDE (2)	Extent of NDE (10)(11)(12)					
D	Tube-to-Tubesheet Welds	See Figure 4.18.13 and Table 4.C.1		MT or PT	100%	100%	100%	100%	25%	10%
E	Permanent attachments (6)	1, 7, 9, 10	With full penetration or partial penetration (15)	RT or UT MT or PT	25% (7) 100%	10% (4) 10%	10% 100%	10% (4) 10%	10% 100%	10% (4) 10% (4)
NA	Pressure retaining areas after removal of attachments	NA		MT or PT	100%	100%	100%	100%	100%	100%
---	Cladding by welding	---		RT or UT	(13)	(13)	(13)	(13)	(13)	(13)
				MT or PT	100%	100%	100%	100%	100%	100%
---	Repairs (14)	---		RT or UT MT or PT	100% 100%	100% 100%	100% 100%	100% 100%	100% 100%	100% 100%

Notes:

1. See paragraph 4.2.
2. RT = Radiographic Examination, UT = Ultrasonic Examination, MT = Magnetic Particle Examination, PT = Liquid Penetrant Examination.
3. 2% if $t \leq$ 30 mm (1-3/16 in) and same weld procedure specification as longitudinal, for steel of P-No.1 Gr 1 and P-No.8 Gr 1
4. 10% if $t >$ 30 mm (1-3/16 in), 0% if $t \leq$ 30 mm (1-3/16 in)
5. Percentage in the table refers to the aggregate weld length of all the nozzles, see paragraph 7.4.3.5 b.
6. RT or UT is not required for weld thicknesses $\leq$ 16 mm (5/8 in)
7. 10% for steel of P-No. 8 Gr 2, P-No 9A Gr 1, P-No 9B Gr 1, P-No. 11A Gr 1, P-No. 11A Gr 2, P-No. 10H Gr 1
8. (Currently not used.)
9. For limitations of application see paragraph 4.2.
10. The percentage of surface examination refers to the percentage of length of the welds both on the inside and the outside.
11. RT and UT are volumetric examination methods, and MT and PT are surface examination methods. Both volumetric and surface examinations are required to be applied the extent shown.
12. NA means "not applicable". All Examination Groups require 100% visual examination to the maximum extent possible.
13. See paragraph 7.4.8.1 for detailed examination requirements.
14. The percentage of examination refers only to the repair weld and the original examination methods, see paragraph 6.2.7.3.
15. RT is applicable only to Type 1, full penetration welds.
16. The term “*a*” as defined in Figure 7.16.
17. The term “*a*” as defined in Figure 7.17.

Table 7.3 – Selection of Nondestructive Testing Method For Full Penetration Joints

Type of Joint	Shell thickness – t	
	$t < 13 mm\ (1/2\ in.)$	$t \geq 13 mm\ (1/2\ in.)$
1, 2, 3	RT	RT or UT per 7.5.5
7, 8	N/A	UT per 7.5.4 or 7.5.5

Table 7.4 – Nondestructive Examination Of Layered Vessels

Joint Category	Weld Joint Description	Type of NDE	Extent
A, B	Category A and B joint in the inner shell and in the inner head	RT or UT	100%
		MT or PT	100%
A	Category A joints in layer 3 mm through 8 mm (1/8 in. through 5/16 in) in thickness	RT or UT	NA
		MT (or PT)	100%
A	Category A joints in layer 8 mm through 16 mm (5/16 in. through 5/8 in) in thickness	UT	10%
		MT (or PT)	100%
A	Category A joints in layer 16 mm through 22 mm (5/8 in. through 7/8 in) in thickness	UT	100%
		MT (or PT)	NA
B	Category B step welded girth joints in layer 3 mm through 8 mm (1/8 in. through 5/16 in) in thickness	RT	SPOT
		MT (or PT)	100%
B	Category B step welded girth joints in layer 8 mm through 16 mm (5/16 in. through 5/8 in) in thickness	RT or UT	NA
		MT (or PT)	100%
B	Category B step welded girth joints in layer 16 mm through 22 mm (5/8 in. through 7/8 in) in thickness	RT	SPOT
		UT	10%
		MT (or PT)	100%
B	Category B step welded girth joints in layer over 22 mm (7/8 in) in thickness	UT	100%
		MT or PT	NA
A, B, D	Category A, B, D full thickness butt welding of solid section to layered section	RT	100%
		MT (or PT)	100%
C	Flat head and tube sheet weld joints of step welded girth joint	Same as Category B step welded girth joint	
D	Nozzle and communicating chamber to layered shell or layered head	MT (or PT)	100%
E	Attachment welds to the pressure boundary	MT (or PT)	100%

Table 7.5 – NDE Techniques, Method, Characterization, Acceptance Criteria

NDT Technique	Method	Paragraph Reference For Characterization And Acceptance Criteria
Visual Examination (VT)	---	7.5.2
Radiographic Examination (RT)	Section V Article 2	7.5.3
Ultrasonic Examination(UT)	Section V Article 4	7.5.4
Ultrasonic Examination (when used in lieu of RT)	Section V Article 4 and paragraph 7.5.5	7.5.5
Magnetic Particle Examination (MT)	Section V Article 7	7.5.6
Liquid Penetrant Examination (PT)	Section V Article 6	7.5.7
Eddy Current Examination (ET)	Paragraph 7.5.8	7.5.8

Table 7.6 – Visual Examination Acceptance Criteria

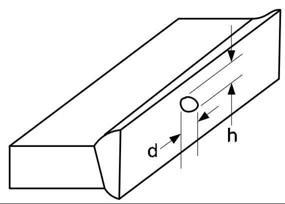
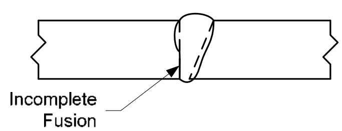
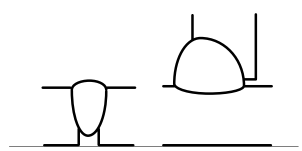
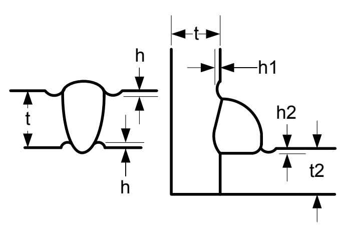
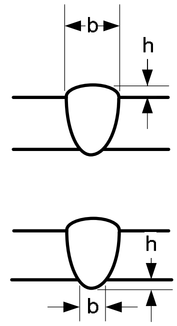
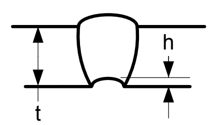
No.	Type Of Imperfection [1]		Acceptance Criteria
1	Cracks (all)	---	Not permitted.
2	Gas cavity (all) Shrinkage cavity (all)		Not Permitted
3	Slag inclusions (all) Flux inclusions (all) Oxide inclusions (all) Metallic inclusions (all)	---	Not permitted when occurring at the surface [2].
4	Incomplete fusion (all)		Not permitted.
5	Lack of penetration		Not permitted if a complete penetration weld is required
6	Undercut		Refer to paragraph 6.2.4.1 (b)(2) for acceptable undercut Requirements in paragraph 7.5.3.2 to permit proper interpretation of radiography shall also be satisfied.

Table 7.6 – Visual Examination Acceptance Criteria

No.	Type Of Imperfection [1]		Acceptance Criteria
7	Weld reinforcement		Acceptable weld reinforcement in butt welding joints shall be in accordance with paragraph 6.2.4.1.d. A smooth transition is required.
8	Joint offset	---	Refer to paragraph 6.1.6 for acceptable offset in butt welded joints.
9	Peaking	---	Refer to paragraph 6.1.6 for acceptable peaking in butt welding joints.
10	Stray flash or arc strike	---	Not permitted [2].
11	Spatter	---	Spatter shall be minimized[2].
12	Torn surface Grinding mark Chipping mark	---	Not permitted[2].
13	Concavity		Refer to paragraph 6.2.4.1(d) for acceptable concavity

Notes:

1) The following symbols are used in this table.
   - a – nominal fillet weld throat thickness
   - b – width of weld reinforcement
   - d – diameter of pore
   - h – height of imperfections
   - t – wall or plate thickness
2) These imperfections may be removed by blend grinding.

Table 7.7 – Radiographic Acceptance Standards For Rounded Indications (Examples Only)

Thickness - t	Maximum Size of Acceptable Rounded Indication		Maximum Size of Non-relevant Indications
	Random	Isolated	
Less than 3 mm (1/8 in)	1/4 t	1/3 t	1/10 t
3 mm (1/8 in)	0.8 mm (1/32 in)	1.1 mm (3/64 in)	0.4 mm (1/64 in)
5 mm (3/16 in)	1.2 mm (3/64 in)	1.5 mm (1/16 in)	0.4 mm (1/64 in)
6 mm (1/4 in)	1.5 mm (1/16 in)	2.1 mm (3/32 in)	0.4 mm (1/64 in)
8 mm (5/16 in)	2.0 mm (5/64 in)	2.6 mm (7/64 in)	0.8 mm (1/32 in)
10 mm (3/8 in)	2.5 mm (3/32 in)	3 mm (1/8 in)	0.8 mm (1/32 in)
11 mm (7/16 in)	2.8 mm (7/64 in)	3.7 mm (5/32 in)	0.8 mm (1/32 in)
13 mm (1/2 in)	3 mm (1/8 in)	4.3 mm (11/64 in)	0.8 mm (1/32 in)
14 mm (9/16 in)	3.6 mm (5/64 in)	5 mm (3/16in.)	0.8 mm (1/32 in)
16 mm (5/8 in)	4.0 mm (5/32 in)	5.3 mm (7/32 in)	0.8 mm (1/32 in)
17 mm (11/16 in)	4.0 mm (5/32 in)	5.8 mm (15.64 in)	0.8 mm (1/32 in)
19 mm (3/4 in) to 50 mm (2 in), inclusive	4.0 mm (5/32 in)	6.4 mm (1/4 in)	0.8 mm (1/32 in)
Over 50 mm (2 in)	4.0 mm (5/32 in)	10 mm (3/8 in)	1.5 mm (1/16 in)

Table 7.8 – Flaw Acceptance Criteria For Welds With A Thickness Between 13 mm (1/2 in.) And Less Than 25 mm (1 in)

Flaw Type	a/t	l
Surface flaw	$\leq$ 0.087	$\leq$ 6.4 mm (1/4 in)
Subsurface flaw	$\leq$ 0.143	$\leq$ 6.4 mm (1/4 in)

Notes:

1) The parameter t is the thickness of the weld excluding any allowable reinforcement, and the parameter l is the length of the flaw. For a butt weld joining two members having different thickness at the weld, t is the thinner of these two thicknesses. If a full penetration weld includes a fillet weld, then the thickness of the throat of the fillet weld shall be included in t .
2) A subsurface indication shall be considered as a surface flaw if the separation (S in Figure 7.11) of the indication from the nearest surface of the component is equal to or less than half the through dimension (2d in Figure 7.11, Sketch (b)) of the subsurface indication.
3) The acceptance limits specified here are based upon workmanship considerations and are not necessarily intended for use in evaluating flaws identified after the vessel has gone into service.
4) *a* and *l* are as defined in paragraph 7.5.5.2.

Table 7.9 – Flaw Acceptance Criteria For Welds With Thickness Between 25 mm (1 in) And Less Than or Equal to 300 mm (12 in)

Flaw Aspect Ratio a/l	25 mm (1 in) $\leq t <$ 64 mm (2 1/2in)		100 mm (4 in) $\leq t \leq$ 300 mm (12 in)	
	Surface Flaw a/t	Subsurface Flaw a/t	Surface Flaw a/t	Subsurface Flaw a/t
0.00	0.031	0.034	0.019	0.020
0.05	0.033	0.038	0.020	0.022
0.10	0.036	0.043	0.022	0.025
0.15	0.041	0.049	0.025	0.029
0.20	0.047	0.057	0.028	0.033
0.25	0.055	0.066	0.033	0.038
0.30	0.064	0.078	0.038	0.044
0.35	0.074	0.090	0.044	0.051
0.40	0.083	0.105	0.050	0.058
0.45	0.085	0.123	0.051	0.067
0.50	0.087	0.143	0.052	0.076

Notes:

1) The parameter t is the thickness of the weld excluding any allowable reinforcement, and the parameter l is the length of the flaw. For a butt weld joining two members having different thickness at the weld, t is the thinner of these two thicknesses. If a full penetration weld includes a fillet weld, then the thickness of the throat of the fillet weld shall be included in t.
2) A subsurface indication shall be considered as a surface flaw if the separation (S in Figure 7.11) of the indication from the nearest surface of the component is equal to or less than half the through dimension (2d in Figure 7.11, Sketch (b)) of the subsurface indication.
3) The acceptance limits specified here are based upon workmanship considerations and are not necessarily intended for use in evaluating flaws identified after the vessel has gone into service.
4) For intermediate flaw aspect ratio a/l and thickness t (64 mm [2 ½ in.] < t < 100 mm [4 in.]), linear interpolation is permissible.
5) If the acceptance criteria in this table results in a flaw length, l, less than 6.4 mm (0.25 in.), a value of 6.4mm (0.25 in.) may be used.
6) For materials exceeding 655 MPa (95 ksi) ultimate tensile strength, the use of this table is limited to a thickness of 200 mm (8 in.).

Table 7.10 – Flaw Acceptance Criteria For Welds With A Thickness Greater Than 300 mm (12 in)

Aspect Ratio a/l	Surface Flaw a	Subsurface Flaw a
0.00	5.8 mm (0.228 in)	6.1 mm (0.240 in)
0.05	6.1 mm (0.240 in)	6.7 mm (0.264 in)
0.10	6.7 mm (0.264 in)	7.6 mm (0.300 in)
0.15	7.6 mm (0.300 in)	8.8 mm (0.348 in)
0.20	8.5 mm (0.336 in)	10.1 mm (0.396 in)
0.25	10.1 mm (0.396 in)	11.6 mm (0.456 in)
0.30	11.6 mm (0.456 in)	13.4 mm (0.528 in)
0.35	13.4 mm (0.528 in)	15.5 mm (0.612 in)
0.40	15.2 mm (0.600 in)	17.7 mm (0.696 in)
0.45	15.5 mm (0.612 in)	20.4 mm (0.804 in)
0.50	15.8 mm (0.624 in)	23.2 mm (0.912 in)

Notes:

1) The parameter t is the thickness of the weld excluding any allowable reinforcement, and the parameter l is the length of the flaw. For a butt weld joining two members having different thickness at the weld, t is the thinner of these two thicknesses. If a full penetration weld includes a fillet weld, then the thickness of the throat of the fillet weld shall be included in t.
2) A subsurface indication shall be considered as a surface flaw if the separation (S in Figure 7.11) of the indication from the nearest surface of the component is equal to or less than half the through dimension (2d in Figure 7.11, Sketch (b)) of the subsurface indication.
3) The acceptance limits specified here are based upon workmanship considerations and are not necessarily intended for use in evaluating flaws identified after the vessel has gone into service.
4) Linear interpolation is permissible for intermediate values of the flaw aspect ratio a/l .
5) This table is not applicable for materials exceeding 655 MPa (95 ksi) ultimate tensile strength.

7.10 Figures

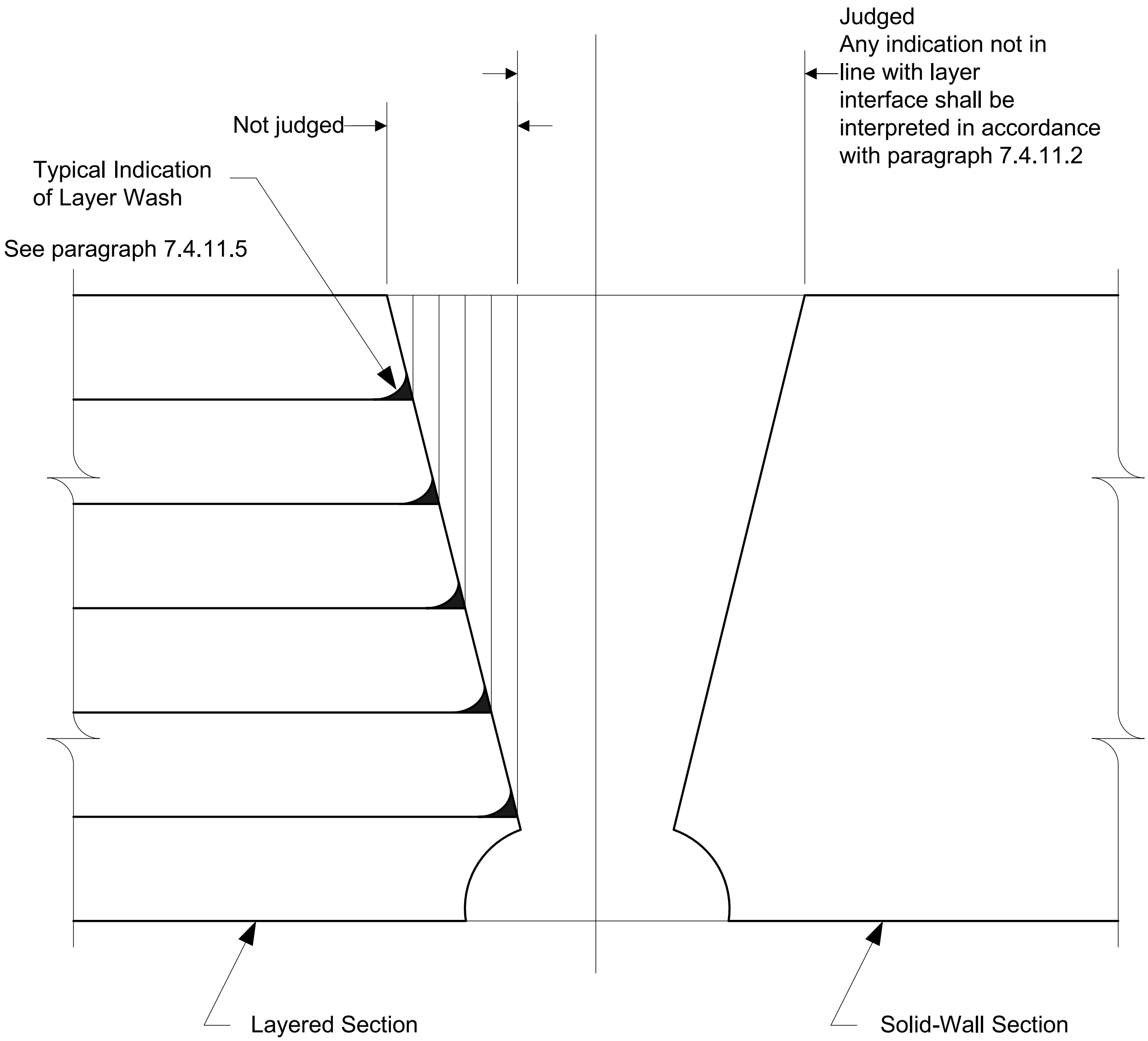


Figure 7.1 – Examination of Layered Vessels

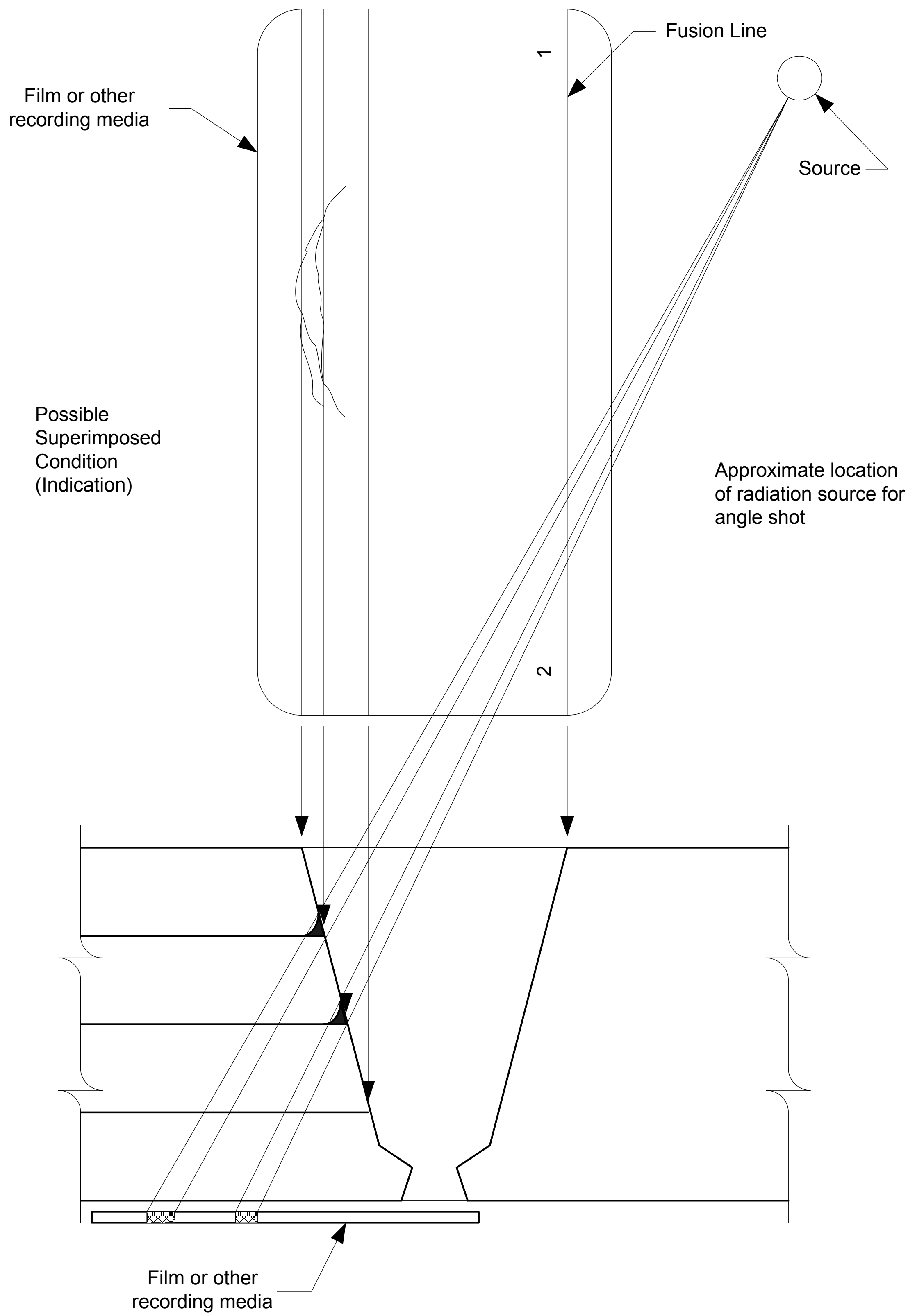
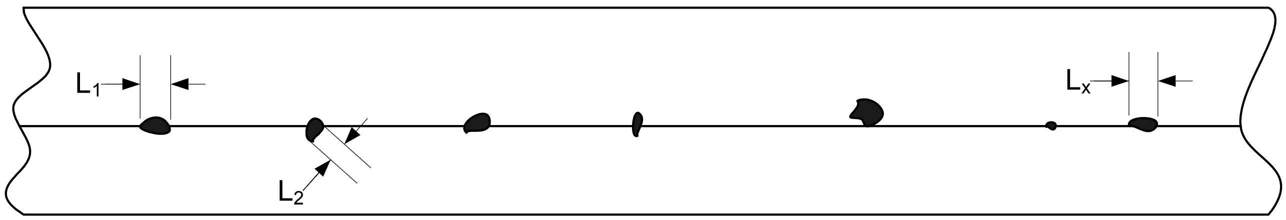
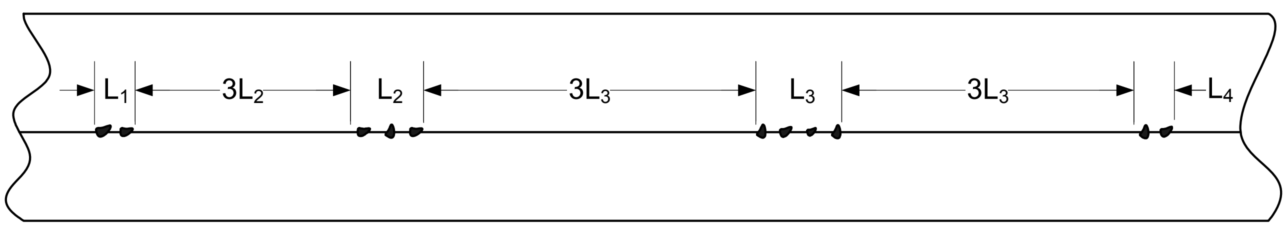


Figure 7.2 – Examination of Layered Vessels



Note: The sum of L_1 to L_x shall be less than t in a length of $12t$.

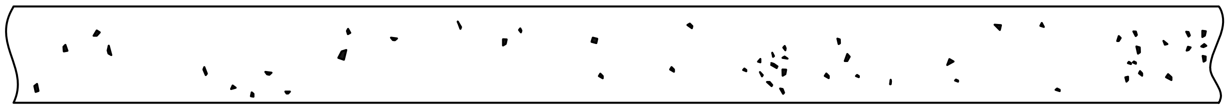
Figure 7.3 – Aligned Rounded Indications



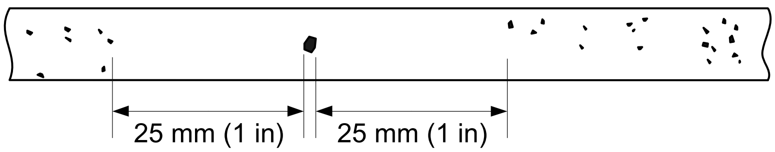
Note: The sum of the group of lengths shall be less than t in a length of $12t$.

Maximum Group Length	Minimum Group Spacing
$L = 6\ mm\ (1/4\ in)$ for $t < 19\ mm\ (3/4\ in)$	$3L$ where L is the length of the longest adjacent group being evaluated
$L = t/3$ for $19\ mm\ (3/4\ in) \le t \le 57\ mm\ (2-1/4\ in)$	
$L = 19\ mm\ (3/4\ in)$ for $t > 57\ mm\ (2-1/4\ in)$	

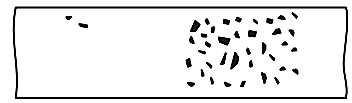
Figure 7.4 – Groups of Aligned Rounded Indications



(a) Random Rounded Indications [See Note (1)]



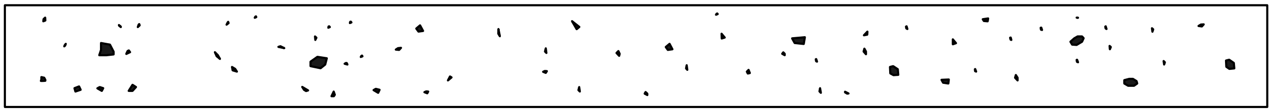
(b) Isolated Indication [See Note (2)]



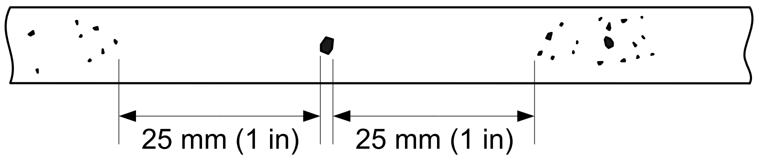
(c) Cluster

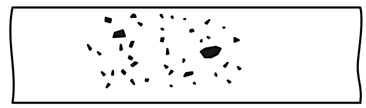
Notes:
1. Typical concentration and size permitted in any 150 mm (6 in) length of weld
2. Maximum size per Table 7.7

Figure 7.5 – Charts for 3 mm (1/8 in) to 6 mm (1/4 in) Wall Thickness, Inclusive



(a) Random Rounded Indications [See Note (1)]



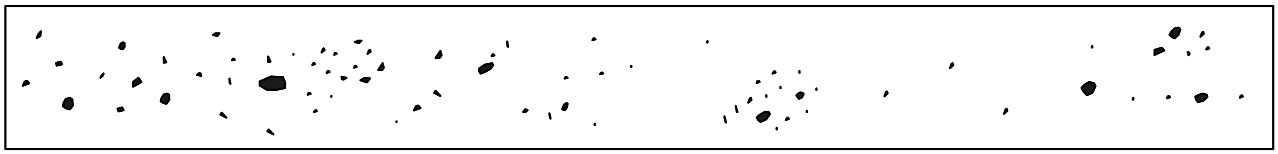


(c) Cluster

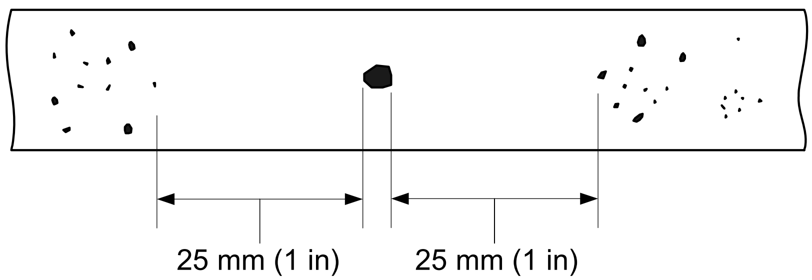
(b) Isolated Indication [See Note (2)]

Notes:
1. Typical concentration and size permitted in any 150 mm (6 in) length of weld
2. Maximum size per Table 7.7

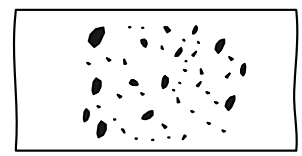
Figure 7.6 – Charts for Over 6 mm (1/4 in) to 10 mm (3/8 in) Wall Thickness, Inclusive



(a) Random Rounded Indications [See Note (1)]



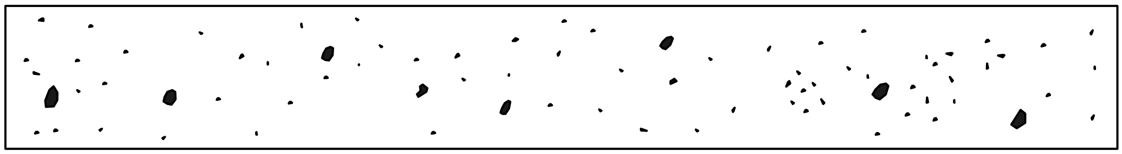
(b) Isolated Indication [See Note (2)]



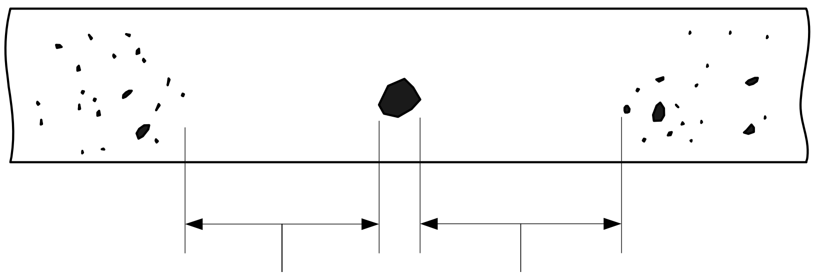
(c) Cluster

Notes:
1. Typical concentration and size permitted in any 150 mm (6 in.) length of weld
2. Maximum size per Table 7.7

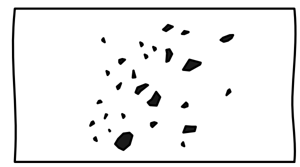
Figure 7.7 – Charts for Over 10 mm (3/8 in) to 19 mm (3/4 in) Wall Thickness, Inclusive



(a) Random Rounded Indications [See Note (1)]



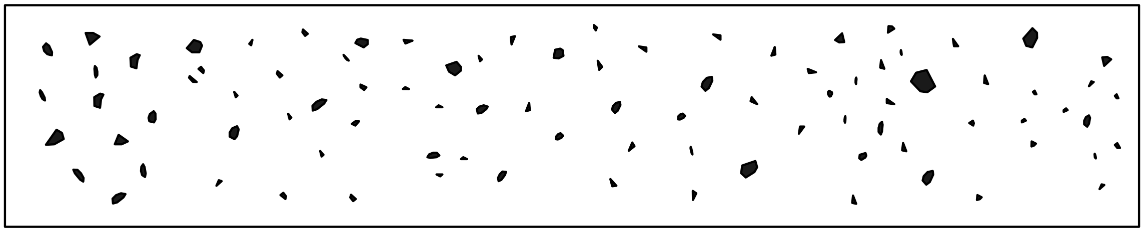
(b) Isolated Indication [See Note (2)]



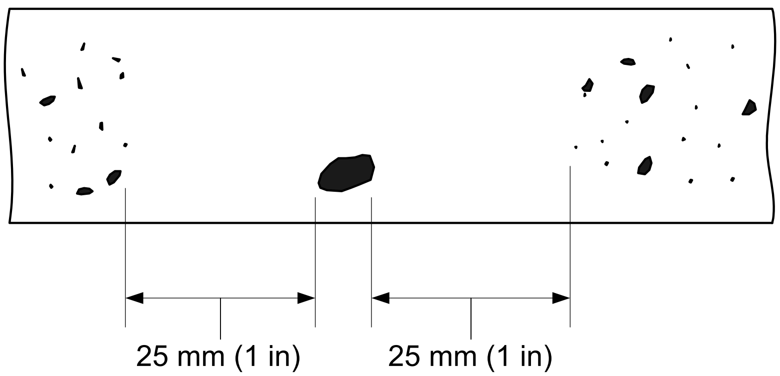
(c) Cluster

Notes:
1. Typical concentration and size permitted in any 150 mm (6 in) length of weld
2. Maximum size per Table 7.7

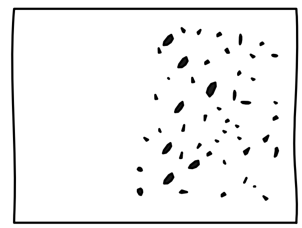
Figure 7.8 – Charts for Over 19 mm (3/4 in) to 50 mm (2 in) Wall Thickness, Inclusive



(a) Random Rounded Indications [See Note (1)]



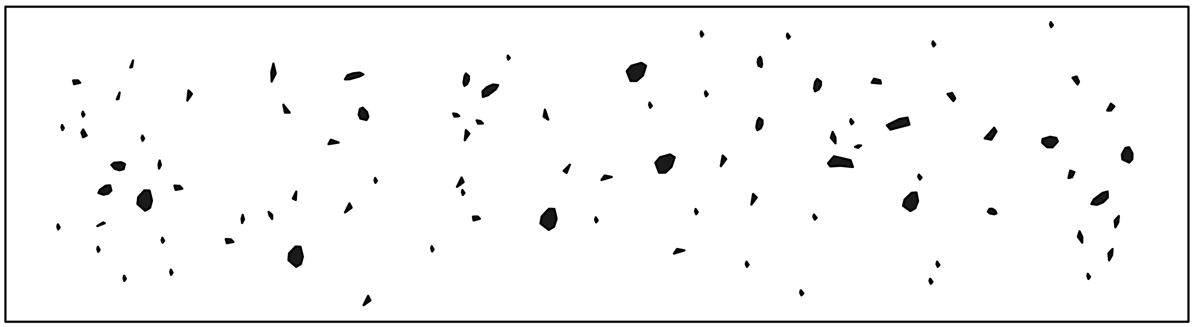
(b) Isolated Indication [See Note (2)]



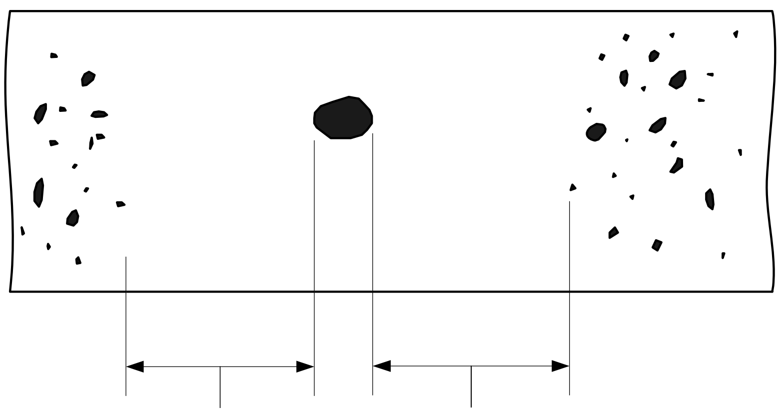
(c) Cluster

Notes:
1. Typical concentration and size permitted in any 150 mm (6 in.) length of weld
2. Maximum size per Table 7.7

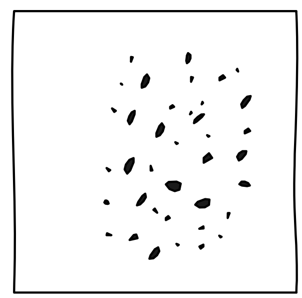
Figure 7.9 – Charts for Over 50 mm (2 in) to 100 mm (4 in) Wall Thickness, Inclusive



(a) Random Rounded Indications [See Note (1)]



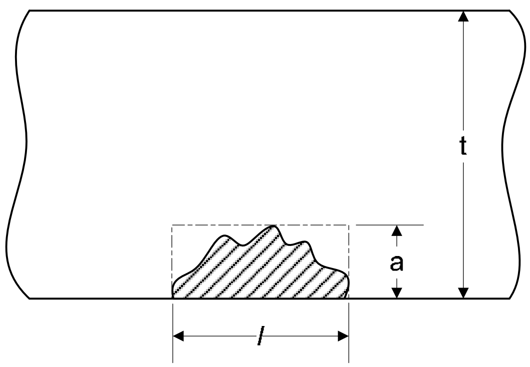
(b) Isolated Indication [See Note (2)]



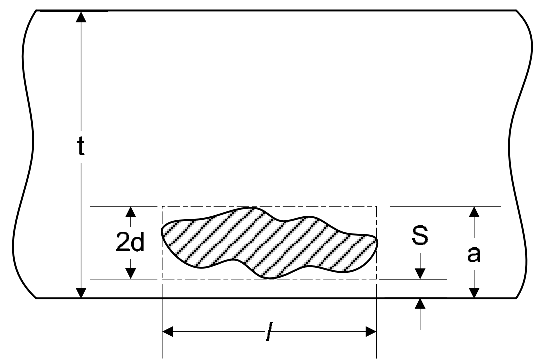
(c) Cluster

Notes:
1. Typical concentration and size permitted in any 150 mm (6 in.) length of weld
2. Maximum size per Table 7.7

Figure 7.10 – Charts for Over 100 mm (4 in.) Wall Thickness

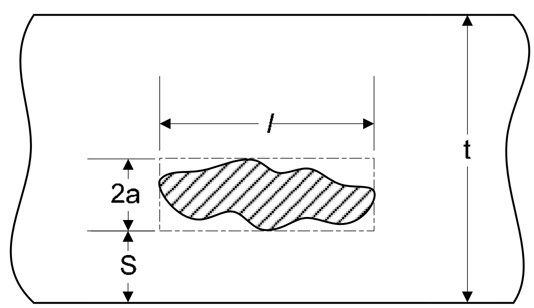


(a) Surface Indication



S ≤ d

(b) Surface Indication



S > a

(c) Subsurface Indications

Figure 7.11 – Single Indications

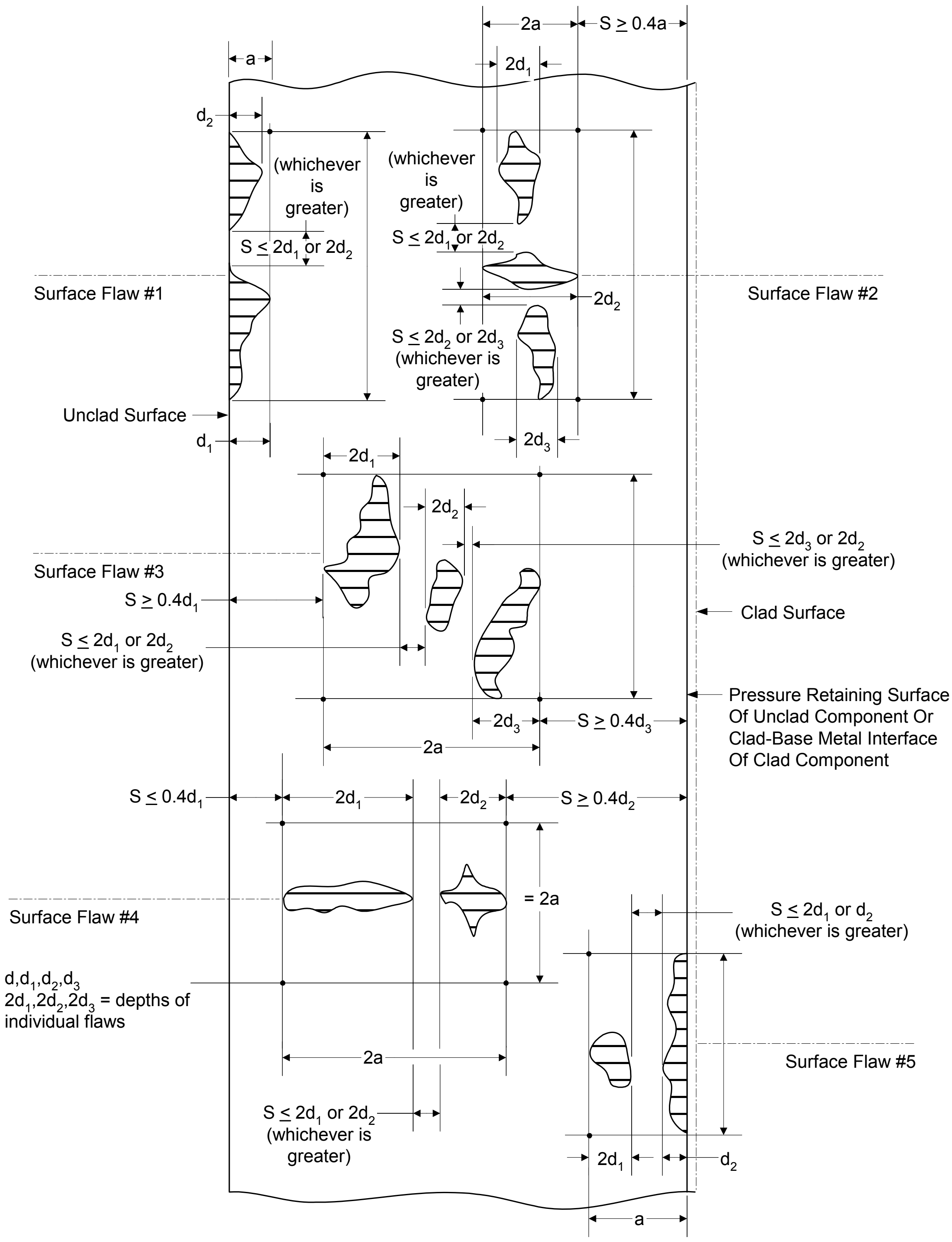


Figure 7.12 – Multiple Planar Flaws Oriented in a Plane Normal to the Pressure Retaining Surface

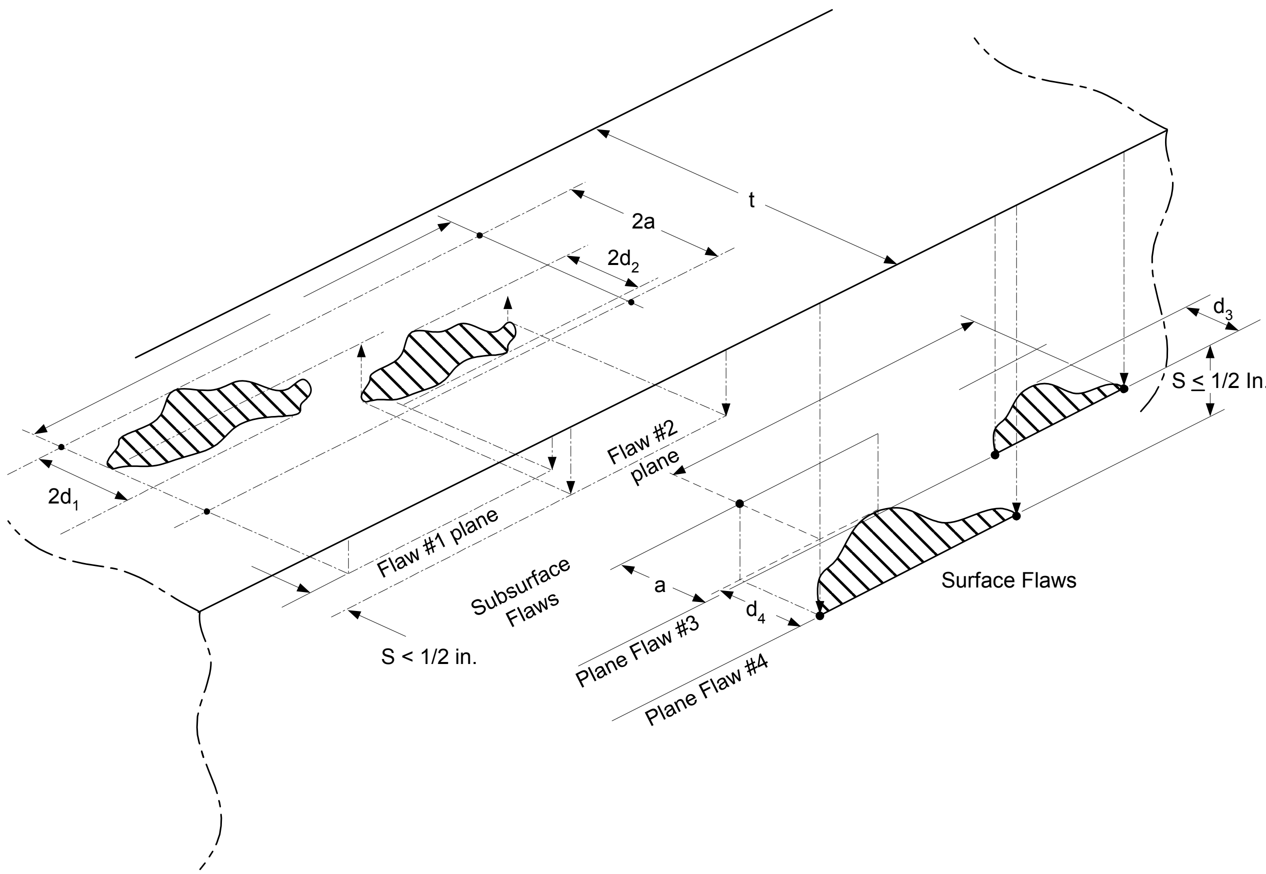


Figure 7.13 – Surface and Subsurface Flaws

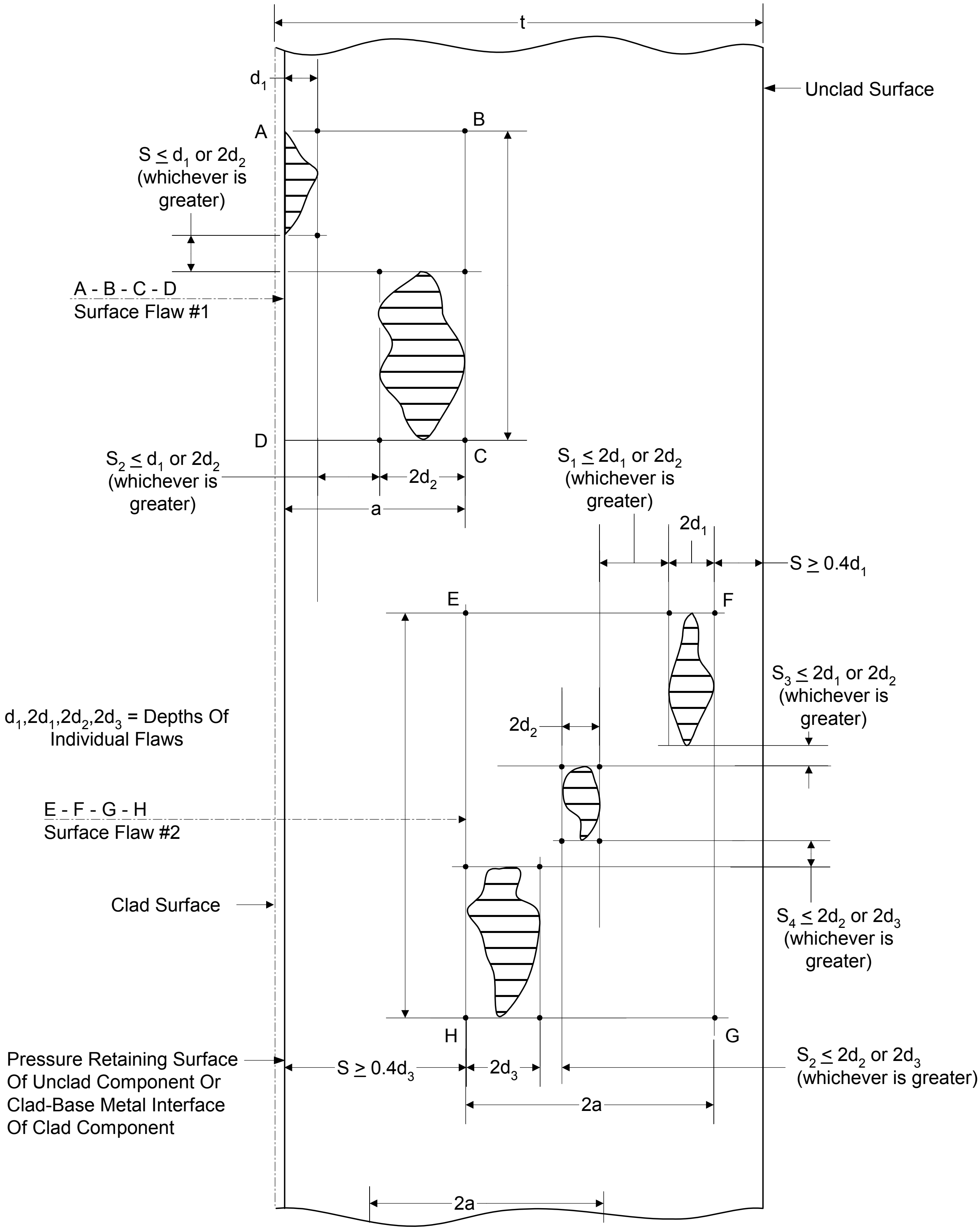
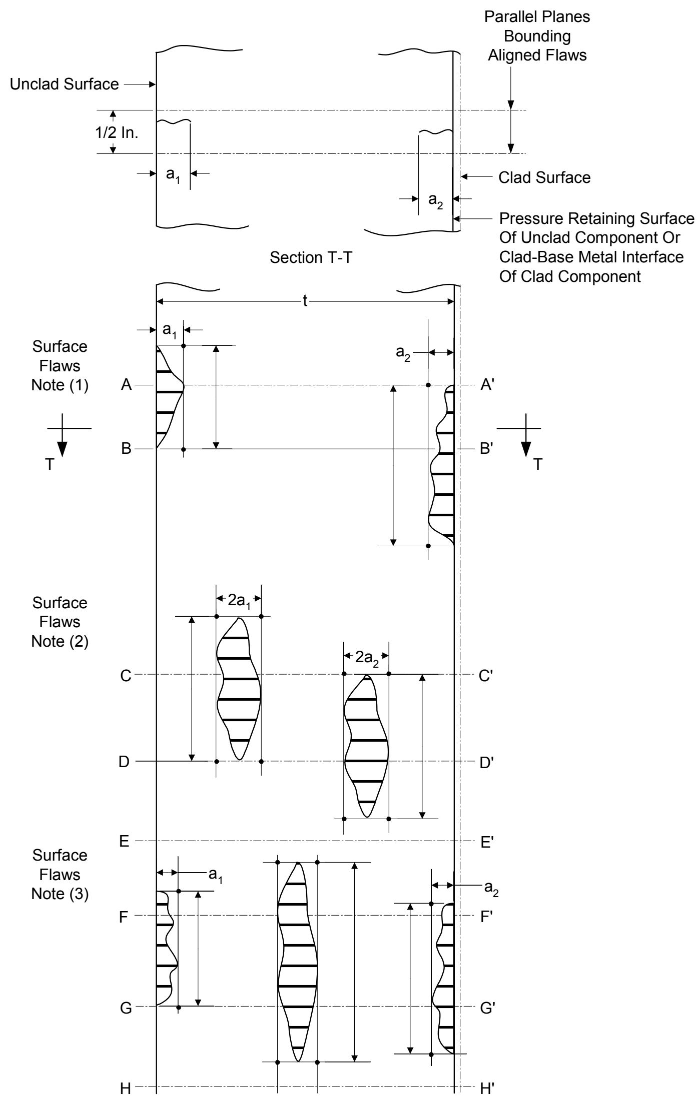


Figure 7.14 – Non-Aligned Coplanar Flaws in a Plane Normal to the Pressure Retaining Surface



Notes:

1) This illustration indicates two surface flaws. The first, a_1, is on the outer surface, and the second, a_2, is on the inner surface: $\left(a_1+a_2\right)\le\left(a_s+a_s^*\right)/2$ within planes $A-A'$ and $B-B'$
2) This illustration indicates two subsurface flaws: $\left(a_1+a_2\right)\le\left(a_e+a_e^*\right)/2$ within planes $C-C'$ and $D-D'$
3) This illustration indicates two surface flaws and one subsurface flaw.

$$\left(a_1+a_3\right)\le\left(a_s+a_e^*\right)/2 \text{ within planes } E-E' \text{ and } F-F'$$

$$\left(a_1+a_2\right)\le\left(a_s+a_e+a_s^*\right)/3 \text{ within planes } F-F' \text{ and } G-G'$$

$$\left(a_2+a_3\right)\le\left(a_s^*+a_e\right)/2 \text{ within planes } G-G' \text{ and } H-H'$$

Figure 7.15 – Multiple Aligned Planar Flaws

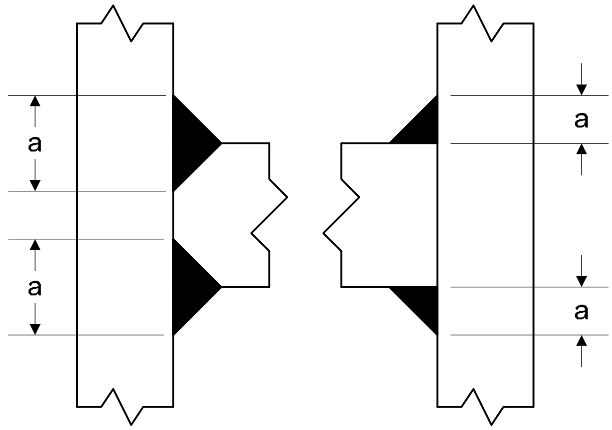


Figure 7.16 - Dimension “a” For Partial Penetration and Fillet Welds

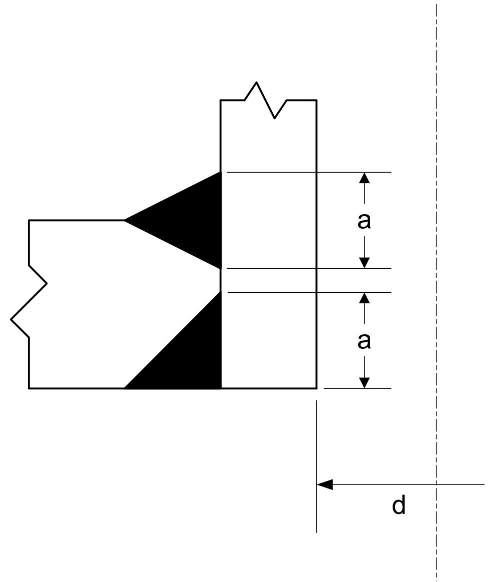


Figure 7.17 - Dimensions “a” and “d” For A Partial Penetration Corner Weld

ANNEX 7.A

RESPONSIBILITIES AND DUTIES FOR INSPECTION AND EXAMINATION ACTIVITIES

(NORMATIVE)

7.A.1 General

The responsibilities and duties for inspection and examination including nondestructive examination during construction of pressure vessels are provided in this Annex. The responsibilities and duties for these activities as related to the specific duties of the Manufacturer and Inspector are covered for vessels to be marked with the Code Symbol.

7.A.2 Manufacturer's Responsibility

7.A.2.1 The Manufacturer

a) The Manufacturer who completes any vessel has the responsibility of complying with all the requirements of this Division and, through proper certification, of ensuring that any work done by others also complies with all requirements of this Division.

b) The Manufacturer has the responsibility of assuring that the quality control, the detailed examinations, and the tests required by this Division are performed at the stages of construction to permit them to be meaningful. The Manufacturer shall provide to the Inspector, at the appropriate time, the information necessary to enable him to perform his specified duties.

7.A.2.2 Inspection and Examination Duties

a) Overview Of Duties – The Manufacturer shall perform his specified duties. Some, but not all of duties pertaining to inspection and examination, which are defined in this Division, that are to be performed by the Manufacturer are summarized in Table 7.A.1.

b) Certification of Competence of Magnetic Particle, Liquid Penetrant, and Eddy Current Examiner – the Manufacturer shall certify that each examiner meets the following requirements.

    1) The examiner has vision, with correction if necessary, to read a Jaeger Type No.2 Standard Chart, at a distance of not less than 300 mm (12 in.), and is capable of distinguishing and differentiating contrast between colors used. Compliance with these requirements shall be demonstrated annually.

    2) The examiner is certified and competent in the techniques of the particular nondestructive examination method, including making the examination, interpretation, and evaluation of the results, except that, where the examination method consists of more than one operation, the examiner may be certified as being qualified only for one or more of these operations.

7.A.3 Inspector's Responsibility

7.A.3.1 The Inspector

a) All references to the Inspectors throughout this Division mean the Authorized Inspector as defined in this paragraph. All inspections required by this Division shall be:

   1) By an Inspector regularly employed by an ASME accredited Authorized Inspection Agency, except that

   2) Inspections may be by the regularly employed user's Inspector in the case of a User-Manufacturer which manufactures pressure vessels exclusively for its own use and not for resale.

   Except as permitted in paragraph 7.A.3.1.a.2, the Inspector shall not be in the employ of the Manufacturer. All Inspectors shall have been qualified by a written examination under the rules of any state of the United States, province of Canada, or any other jurisdiction that has adopted the Code.

b) Whenever Authorized Inspection Agency or AIA is used in this Division, it shall mean an Authorized Inspection Agency accredited by ASME in accordance with the requirements in the latest edition of ASME QAI-1, Qualifications for Authorized Inspection.

7.A.3.2 Inspection and Examination Duties

7.A.3.2.1 General

a) The Inspector shall make all inspections specifically required by the rules of this Division plus such other inspections the Inspector believes that are necessary to enable the Inspector to certify that the vessel to be stamped with the Code Symbol has been designed and constructed in accordance with the requirements of this Division.

b) Some, but not all, of the required inspections and verifications that are defined in the rules of this Division are summarized in Table 7.A.1.

7.A.3.2.2 Manufacturer's Quality Control System

In addition to the duties specified, the Inspector has the duty to monitor the Manufacturer's Quality Control System.

7.A.3.2.3 Inspection of Materials

a) Compliance Of Materials With Requirements – the Inspector shall assure himself that all materials used comply in all respects with the requirements of this Division. The Manufacturer shall submit to the Inspector certification of materials compliance. The Inspector shall examine certified test reports or certificates of compliance for the materials used, except as otherwise provided for in the material specification or in this Division.

b) Marking On Materials – the Inspector shall inspect materials used in the construction to see that they bear the identification required by the applicable material specification, except as otherwise provided in Part 3 of this Division. Should the identifying marks be obliterated or the material be divided into two or more parts, the marks shall be properly transferred by the Manufacturer as provided in paragraph 6.1.1.2.

7.A.3.2.4 Dimensional Check of Component Parts

a) The Inspector shall satisfy himself that:

1) Head and shell sections conform to the prescribed shape and meet the thickness requirements after forming;

2) Nozzles, manhole frames, reinforcement around openings, and other appurtenances to be attached to the inside or outside of the vessel fit properly to the curvature of the vessel surface; and

3) The dimensional requirements have been met, include making such dimensional measurements as the Inspector considers necessary.

b) Use of Templates **–** if required by the Inspector, the Manufacturer of the vessel shall furnish accurately formed templates for his use.

7.A.3.2.5 Check Of Heat Treatment Practice

The Inspector shall satisfy himself that the Manufacturer has conducted all heat treatment operations required by this Division. Certificates furnished by the Manufacturer may be accepted as evidence that the heat treatment operations were correctly carried out.

7.A.3.2.6 Inspection of Welding

a) Check Of Welding Procedure Specifications – the Inspector shall verify that the welding procedures employed in construction have been qualified under the provisions of Section IX and as specified in this Division. The Manufacturer shall submit evidence to the Inspector that those requirements have been met. When there is a specific reason to question a welding procedure, the Inspector may require re-qualification as a requirement for the procedure to be used on work subject to his inspection.

b) Check of Welder and Welding Operator Performance Qualification – the Inspector shall verify that all welding is done by welders or welding operators qualified under the provisions of Section IX. The Manufacturer shall make available to the Inspector a certified copy of the record of performance qualification tests of each welder and welding operator as evidence that these requirements have been met. When there is a specific reason to question the ability of a welder or welding operator to make welds that meet the requirements of the Welding Procedure Specification, the Inspector may require re-qualification as a requirement for the welder or welding operator to continue welding on work subject to his inspection.

c) Check Of Nondestructive Examination Methods – the Inspector shall verify that the nondestructive examination methods of Part 7 which are used follow the techniques specified therein, that the examinations are performed by operators who are certified by the Manufacturer as being qualified in the techniques of the methods employed and in the interpretation and evaluation of the results, and that the Manufacturer has met the requirements of all of the rules of this Division. If there is a specific reason to question an operator's qualifications, the Inspector has the right to require proof of the operator's ability to perform and interpret the examinations specified. The Inspector may witness nondestructive examinations at his discretion.

7.A.3.2.7 Witness of Pressure Test

The Inspector shall witness the hydrostatic test or pneumatic test required by Part 8 of this Division.

7.A.4 Tables

Table 7.A.1 – Inspection And Examination Activities and Responsibilities/Duties

Inspection And Examination Activities	Time of Examination	Paragraph Reference		Manufacturer's Responsibilities	Inspector's Duties
		Procedure	Acceptance Criteria		
The Certificate Of Authorization From ASME Boiler And Pressure Vessel Committee	Before Start Of All Work	Annex 2.G Annex 2.E	NA	Obtain the Certificate and maintain Quality Control System	Verify the validity of Certificate and that Quality Control System is in place and being followed.
Manufacturers Quality Control System		2.3.5 Annex 2.E	7.A.3.2.2	Maintain and Quality Control System	Verify that Quality Control System is in place, Monitor the Quality Control System During Fabrication
The Applicable Drawings And Documents	Before Fabrication	NA	2.2.2 Part 4 Part 5	Prepare applicable Design Report, User Design Specification (if applicable), drawings and related documents	Verify that applicable Design Report, User Design Specification, drawings and related documents are available
Compliance Of All Material Used In The Fabrication Of The Vessel Or Part Including Sample Test Coupons	Before Fabrication	7.3.1.1	Part 3 7.A.3.2.3	Make certain that material used comply with the requirements of Part 3	Verify compliance of material with the requirements of Part 3
Repair Of Material Defects	Before Fabrication	6.1.1.3	6.1.1.3	Make certain that material defects repaired by welding are acceptably repaired and reexamined	Verify that material defects repaired by welding are acceptably repaired and reexamined
Traceability Of The Material Identification	During Cutting Of Material	7.3.1.2 3.2.7.2	NA	Make certain that the material identification numbers have been properly transferred	Make examinations to confirm that the material identification numbers have been properly transferred
Proper Thickness And Dimensional Check Of Vessel Components	Before Welding	7.3.2	6.2 7.A.3.2.4	Examine to confirm they have been properly formed to shape within tolerances	Verify that the thickness and dimensions are within tolerances
Qualification Of Welding Procedure	Before Welding	6.2.2.4 Sec.IX	7.3.4 7.A.3.2.6.a Sec.IX	Perform and maintain qualification	Verify that all welding procedures have been qualified
Qualification Of Welders And Welding Operators	Before Welding	6.2.2.5 Sec.IX	7.3.5 7.A.3.2.6.b Sec.IX	Perform and maintain qualification	Verify that all welders and welding operators have been qualified
Repair of Material Cut Edge Defects	During Fabrication	6.1.3.1	7.4.4 7.4.5 7.4.6	Make certain that material edge defects repaired by welding are acceptably repaired and reexamined	Verify that material cut edge defects repaired by welding are acceptably repaired and reexamined

Table 7.A.1 – Inspection And Examination Activities and Responsibilities/Duties - Continued

Inspection And Examination Activities	Time of Examination	Paragraph Reference		Manufacturer's Responsibilities	Inspector's Duties
		Procedure	Acceptance Criteria		
Proper Fitting And Cleaning Of Parts For Welding	Before Welding	6.1.3 6.1.4 6.1.5 6.1.6	7.3.6	Examine all parts to make certain they have been properly fitted/aligned and the surfaces to be joined have been cleaned for welding	Verify that all parts have been properly fitted/aligned and the surfaces to be jointed have been cleaned for welding
Any Repairs For Defects By Welding	During Fabrication	6.2.7	7.4.2 through 7.4.6	Make certain that weld defects are acceptably repaired and reexamined	Verify that weld defects are acceptably repaired and reexamined
Control For Required Heat Treatments	During Fabrication	6.4	7.3.3 7.A.3.2.5	Control to assure that all required heat treatments are performed	Verify that the heat treatments, including PWHT have been performed properly
Impact Tests For Welds As Production Test	After Welding	3.11.8	3.11.8	Perform tests and provide records	Verify that impact tests have been performed and that the results are acceptable
Certification of Qualification of Nondestructive Radiographic, Ultrasonic, Magnetic Particle, Liquid Penetrant, and Eddy Current Test Examiners	After Welding	7.3.6	7.A.3.2.6.c	Certify that each operator meets requirements of this Division	Verify that each operator meets requirements of the Division
Nondestructive Examinations	After Welding	7.4 7.A.3.2.6.c	7.4.3 7.4.4 7.4.5 7.4.6 7.4.7	Perform examinations and provide records including retaining radiographs and UT Scans	Verify that required nondestructive examinations have been performed and that the results are acceptable
Visual Examinations	After Welding	7.5.2	Table 7.6	Perform visual examinations	Make a visual inspection of the vessel to confirm that there are no welding and dimensional defects
Hydrostatic Or Pneumatic Test With Required Inspection During Such Test	After Fabrication	Part 8	Part 8, 7.A.3.2.7	Perform inspection and test	Perform inspections and witnessing the hydrostatic or pneumatic tests
Stamping And/Or Nameplate To The Vessel	After Fabrication	Annex 2.F	NA	Apply the required stamping and/or nameplate to the vessel	Verify that the required marking, including stamping, is provided and that any name plate has been attached
Manufacturer's Data Report	After Fabrication	Annex 2.D	NA	Prepare, certify, and provide to the Inspector for certification	Sign the Certificate of inspection
Manufacturers Data Report and Records Specified by this Division	After Delivery	Annex 2.C	NA	Maintain proper records and distribute the documentation package	Verify that the Manufacturer has maintained proper records during Vessel Manufacture

PART 8

PRESSURE TESTING REQUIREMENTS

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8.1 General Requirements

8.1.1 Selection of Pressure Test Methods

a) Except as otherwise permitted in paragraphs 8.1.1.b and 8.1.1.c, a completed vessel designed for internal pressure shall be subjected to a hydrostatic test performed in accordance with paragraph 8.2. Pressure tests of vessels designed for vacuum or partial vacuum only shall be tested in accordance with paragraph 8.1.3.1. A vessel shall be considered a completed vessel after:

  1) All fabrication has been completed, except for operations that could not be performed prior to the test such as weld end preparation, or cosmetic grinding on the base material that does not affect the required thickness including corrosion allowance.

  2) All examinations have been performed, except those required after the test.

b) Subject to the limitations and additional nondestructive weld examination requirements that may be imposed elsewhere in this Division, a pneumatic test performed in accordance with paragraph 8.3 may be substituted for a hydrostatic test if any of the following are true.

  1) The vessel is constructed and supported such that the weight of the hydrostatic test fluid could cause permanent visible distortion.

  2) The vessel cannot be readily dried and is to be used in services where traces of the testing liquid cannot be tolerated.

  3) The vessel is so constructed that brittle fracture is not a credible mode of failure at the pressure test conditions.

  4) The pneumatic test is monitored by acoustic emission examination in accordance with Article 12 of Section V.

c) Combined hydrostatic-pneumatic tests may be substituted in cases where it is desirable to test a vessel partially filled with liquid. Combined hydrostatic-pneumatic tests shall be performed in accordance with paragraph 8.4.1.

8.1.2 Precautions

a) Pressure tests shall be carried out under controlled conditions with appropriate safety precautions and equipment.

b) Vents shall be provided at all high points of the vessel in the position in which it is to be tested to allow purging possible air pocket locations while the vessel is filled for hydrostatic testing. Attention shall be given to nozzle protrusions and vessel internals.

c) When performing a pneumatic test, particular care shall be taken to avoid brittle fracture given the potential hazards of the energy stored in the compressed gas. In this regard, the decision to perform a pneumatic test shall be considered during the design of the vessel so that the minimum design temperature/coincident pressure conditions for all pressure-boundary components, including any reduction in temperature and to a coincident reduction in pressure of the service fluid as the design pressure is released (auto-refrigeration), are considered when selecting the materials of construction.

d) Air or gas is hazardous when used as a testing medium. It is therefore recommended that the vessel be tested in such a manner as to ensure personnel safety from a release of the total internal energy of the vessel. See also ASME PCC-2, Article 5.1, Appendix III "Safe Distance Calculations for Pneumatic Pressure Test" and Appendix II "Stored Energy Calculations for Pneumatic Pressure Test." Liquid test media may also present hazards due to the stored energy in the compressed liquid and strain energy stored in the vessel material.

e) Vessels may be painted or coated either internally or externally, and may be lined internally, prior to pressure testing. However, the application of paints, coatings and linings is not permitted prior to hydrotest if the vessel is to contain fluids of such a nature that a very small amount mixed or unmixed

with air is dangerous to life. The user is cautioned that the application of paints, coatings, or linings may mask leaks that would otherwise be detected during the pressure test.

8.1.3 Requirements for Vessels of Specific Construction

8.1.3.1 Vessels Designed for Vacuum or Partial Vacuum Only

Vessels designed for vacuum or partial vacuum only and chambers of multi-chamber vessels designed for vacuum or partial vacuum only shall be subjected to a pressure test in accordance with paragraph 8.1.1. The internal test pressure shall not be less than 1.43 times the difference between normal atmospheric pressure and the minimum design internal absolute pressure.

8.1.3.2 Jacketed Vessels

a) For jacketed portions of vessels where the internal vessel is designed to operate at atmospheric pressure or vacuum conditions only, the pressure test need only be applied to the jacket volume. In such cases, the MAWP shall be set as the differential pressure between the jacket and the internal vessel for the purposes of determining the test pressure.

b) If the jacket is designed to operate under vacuum conditions, it shall be tested in accordance with paragraph 8.1.3.1.

c) If the jacket is designed to operate under both pressure and vacuum conditions, then it shall be tested at the greater of the pressures determined in accordance with paragraph 8.1.3.2.a or 8.1.3.2.b.

8.1.3.3 Combination Units

Combination units shall be tested by one of the following methods

a) Pressure chambers of combination units that have been designed to operate independently shall be hydrostatically tested as separate vessels; that is, each chamber shall be tested without pressure in the adjacent chamber. If the common elements of a combination unit are designed for a larger differential design pressure than the higher maximum allowable working pressure to be marked on the adjacent chambers, the hydrostatic test shall subject the common elements to at least their design differential pressure, corrected for temperature as described in paragraph 8.2.1.b, as well as meet the requirements of paragraph 8.2.1.a or 8.2.1.e for each independent chamber.

b) When pressure chambers of combination units have their common elements designed for the maximum differential pressure that can possibly occur during startup, operation (including upset conditions) and shutdown, and the differential pressure is less than the higher pressure in the adjacent chambers, then the common elements shall be subjected to a hydrostatic test pressure calculated using Equation (8.2) where the MAWP is the differential pressure to be marked on the unit.

    1) Following the test of common elements as required in paragraph 8.1.3.3.a, and their inspection, the adjacent chambers shall be simultaneously tested at the test pressure required for internal pressure. Care must be taken to limit the differential pressure between the chambers to the pressure used when testing common elements.

    2) The vessel stamping and vessel Data Report shall describe the common elements and their limiting differential pressure.

8.1.3.4 Lined Vessels

a) For lined vessels, a test is recommended for the pressure tightness of the applied lining that is appropriate for the intended service. Details of the test shall be a matter for agreement between the user and the Manufacturer. The test should be such as to ensure freedom from damage to the load-carrying base material. When corrosion of the base material is to be expected from contact with the contents of the vessel, particular care should be taken in devising and executing the tightness test.

b) Following the hydrostatic pressure test, the interior of the vessel shall be inspected to determine if there is any seepage of the test fluid through the joints in the lining.

c) When the test fluid seeps behind the applied liner, there is danger that the fluid will remain in place until the vessel is put in service. In cases where the operating temperature of the vessel is above the boiling point of the test fluid, the vessel should be heated slowly for a sufficient time to drive out all test fluid from behind the applied liner without damage to the liner. This heating operation shall be performed at the vessel manufacturing plant. After the test fluid is driven out, the lining should be repaired as required. Repetition of the radiography, the heat treatment, or the hydrostatic test of the vessel after lining repairs is not required except when there is reason to suspect that the repair welds may have defects that penetrate into the base material, in which case an Inspector shall decide which one or more shall be repeated.

d) As an alternative to the procedure in paragraph 8.1.3.5.c, it is recommended that consideration be given to adding a weep hole at a low point in each pressure boundary component that is protected by a liner panel that is seal welded all around the panel to the pressure boundary component. These weep holes should be monitored for leakage during both testing and operation and will minimize pressure build-up behind the panels, a circumstance that could cause the panel to buckle upon release of the internal pressure in the vessel.

8.1.3.5 Layered Vessels

Pneumatic testing is not permitted when using the procedures of paragraph 4.13.12.2 to measure the contact between layers during construction.

8.1.4 Pressure Gages

a) Pressure gages used in testing vessels shall be indicating pressure gages and shall be connected directly to the vessel. If the indicating gage is not readily visible to the operator controlling the pressure applied from a safe location, an additional indicating gage shall be provided where it will be visible to the operator and Inspector throughout the duration of the test. It is recommended that a recording gage be used in addition to the indicating gage.

b) Dial indicating pressure gages used in testing shall be graduated over a range of about two times the maximum intended test pressure, but in no case shall the range be less than one and one-half times nor more than four times the intended test pressure. Digital reading pressure gages having a wider range may be used provided the readings give the same or a greater degree of accuracy than obtained with dial pressure gages.

c) All gages shall be calibrated against a standard deadweight tester or a calibrated master gage at least every 6 months or at any time there is a reason to believe that they are in error.

8.2 Hydrostatic Testing

8.2.1 Test Pressure

a) Except as noted for vessels of specific construction identified in paragraph 8.1.3, the minimum hydrostatic test pressure shall be the greater of:

$$P_T = 1.43 \cdot MAWP \tag{8.1}$$

or

$$P_T = 1.25 \cdot MAWP \cdot \left(\frac{S_T}{S}\right) \tag{8.2}$$

b) The ratio S_T/S in Equation (8.2) shall be the lowest ratio for the pressure-boundary materials, excluding bolting materials, of which the vessel is constructed.

c) The test pressure is the pressure to be applied at the top of the vessel during the test. This pressure plus hydrostatic head is used in the applicable design equations to check the vessel under test conditions, see Part 4, paragraph 4.1.6.2.a.

d) The requirement of paragraph 8.2.1.a represents the minimum required hydrostatic test pressure. The upper limits of the test pressure shall be determined using the method in paragraph 4.1.6.2.a. Any intermediate value of pressure may be used.

e) A hydrostatic test based on a calculated pressure may be used by agreement between the user and the Manufacturer. The hydrostatic test pressure at the top of the vessel shall be the minimum of the test pressures calculated by multiplying the basis for the calculated test pressure for each pressure element by 1.43 and reducing this value by the hydrostatic head on that element. The basis for this calculated test pressure is the highest permissible internal pressure, as determined by the design equations, for each element of the vessel using the nominal thicknesses, including corrosion allowance, and the allowable stress values given in Part 3, Annex 3.A for the temperature of the test. When this pressure is used, it shall be as set forth in the Manufacturer's Design Report.

8.2.2 Preparation for Testing

Before applying test pressure, the test equipment shall be inspected to see that it is tight and that all low-pressure filling lines and other appurtenances that should not be subjected to the test pressure have been disconnected or isolated by valves or other suitable means.

8.2.3 Test Fluid

Any liquid, non–hazardous at any temperature, may be used for hydrostatic testing if below its boiling point. Combustible liquids having a flash point less than 45°C (110°F) such as petroleum distillates, may be used only for atmospheric temperature tests.

8.2.4 Test Procedures

a) The metal temperature during a hydrostatic test shall be maintained at least 17°C (30°F) above the minimum design metal temperature of the vessel, but need not exceed 50°C (120°F), to minimize the risk of brittle fracture.

b) The test pressure shall not be applied until the vessel and the test fluid are at about the same temperature.

c) Hydrostatic pressure shall be gradually increased until the test pressure is reached. The pressure shall then be reduced to a value not less than the test pressure divided by 1.43 before examining for leakage in accordance with paragraph 8.2.5.

8.2.5 Test Examination and Acceptance Criteria

a) Following the reduction of the test pressure to the level indicated in paragraph 8.2.4.c, a visual examination for leakage shall be made by the Inspector of all joints and connections and of all regions of high stress such as knuckles of formed heads, cone-to-cylinder junctions, regions around openings, and thickness transitions. Visual examination of the vessel may be waived provided all of the following requirements are satisfied:

    1) A suitable gas leak test is applied, see paragraph 8.4.2.

    2) Substitution of the gas leak test is by agreement between the Manufacturer and Inspector.

    3) All welded seams that will be hidden by assembly are given a visual examination for workmanship prior to assembly.

b) Any leaks that are present, except for that leakage that may occur at temporary test closures for those openings intended for welded connections, shall be corrected and the vessel shall be retested.

c) The Inspector shall reserve the right to reject the vessel if there are any visible signs of permanent distortion.

8.3 Pneumatic Testing

8.3.1 Test Pressure

a) Except for enameled vessels whose test pressure shall be at least the MAWP to be marked on the vessel, the minimum pneumatic test pressure shall be computed from Equation (8.3).

$$P_T = 1.15 \cdot MAWP \cdot \left(\frac{S_T}{S}\right) \tag{8.3}$$

b) The ratio S_T/S in Equation (8.3) shall be the lowest ratio for the pressure-boundary materials, except bolting materials, of which the vessel is constructed.

c) The requirements of paragraph 8.3.1.a represent the minimum required pneumatic test pressure required by this Division. The upper limits of this test pressure can be determined using the method in Part 4, paragraph 4.1.6.2.b. Any intermediate value may be used.

8.3.2 Preparation for Testing

Prior to testing, test equipment shall be examined to ensure that it is tight and all filling lines and other appurtenances that should not be subjected to the test pressure have been disconnected or isolated by valves or other suitable means.

8.3.3 Test Fluid

Any pressurizing medium used in pneumatic testing shall be nonflammable and nontoxic. When compressed air is used for a pressure test, the following should be considered:

a) Use only clean, dry, oil-free air meeting the requirements of Class 1, 2, or 3 air per ISO 8573-1.

b) The dew point of the air should be between -20 ^{o}C to -70 ^{o}C (-4 ^{o}F to -94 ^{o}F).

c) Verification that there is no hydrocarbon contamination or other organic residue within the vessel since this could result in the formation of an explosive mixture.

8.3.4 Test Procedures

a) The metal temperature during a pneumatic test shall be maintained at least 17^{o}C (30^{o}F) above the minimum design metal temperature to minimize the risk of brittle fracture.

b) The test pressure shall not be applied until the vessel and the test fluid are at about the same temperature.

c) Test pressure shall be gradually increased until one-half of the test pressure is reached after which the test pressure shall be increased in steps of approximately one-tenth of the test pressure until the test pressure has been reached. The pressure shall then be reduced to a value not less than the test pressure divided by 1.15 before examining for leakage in accordance with paragraph 8.3.5.

8.3.5 Test Examination and Acceptance Criteria

a) Following the reduction of the test pressure to the level indicated in paragraph 8.3.4.c, a visual examination for leakage shall be made. The Inspector shall witness this examination. Except for leakage that might occur at temporary test closures for those openings intended for welded connections, leakage is not allowed at the time of the required visual inspection. Leakage from temporary seals shall be directed away so as to avoid masking leaks from other joints. Visual examination of the vessel may be waived provided:

   1) a suitable gas leak test is applied, see paragraph 8.4.2,

   2) substitution of the gas leak test is by agreement between the Manufacturer and Inspector,

3) all welded seams that will be hidden by assembly are given a visual examination for workmanship prior to assembly.

b) Any leaks that are present, except for that leakage that may occur at temporary test closures for those openings intended for welded connections, shall be corrected and the vessel shall be retested.

c) The Inspector shall reserve the right to reject the vessel if there are any visible signs of permanent distortion.

8.4 Alternative Pressure Testing

8.4.1 Hydrostatic–Pneumatic Tests

In cases where it is desirable to pressure test a vessel partially filled with liquid, the requirements of paragraph 8.3 shall be met, except the pneumatic pressure applied above the liquid level shall at no point result in a total pressure that causes the general membrane stress to exceed 80% of the specified minimum yield strength of the material at test temperature.

8.4.2 Leak Tightness Testing

a) Leak tightness tests include a variety of methods of sufficient sensitivity to allow for the detection of leaks in pressure elements, including, but not limited to the use of direct pressure and vacuum bubble test methods, and various gas detection tests.

b) The selection of a leak tightness test to be employed should be based on the suitability of the test for the particular pressure element being tested.

c) The metal temperature for leak tightness tests shall be in accordance with paragraph 8.3.4.a. Additionally, the temperature shall be maintained within the specified range for the test equipment being used.

d) Leak tightness tests shall be performed in accordance with Article 10 of Section V.

8.5 Documentation

For all pressure tests, as a minimum, the following data shall be recorded by the Manufacturer and shall be issued as part of the vessel's Data Report:

a) Vessel Manufacturer and identification of the pressure vessel

b) Name of Authorized Inspection Agency

c) Type of test (hydrostatic, pneumatic, hydrostatic–pneumatic)

d) Test pressure at the top of the vessel in the test position

e) Position of the vessel (horizontal, vertical, normal operating)

f) Test fluid and temperature

g) Date of pressure test

h) If a written pressure test procedure is followed, reference shall be made to this procedure.

8.6 Nomenclature

$MAWP$	maximum allowable working pressure.
P_T	minimum test pressure.
S	allowable stress from Annex 3.A. evaluated at the design temperature.
S_T	allowable stress from Annex 3.A evaluated at the test temperature.

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PART 9

PRESSURE VESSEL OVERPRESSURE PROTECTION

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9.1 General Requirements

a) This Part provides the requirements for pressure relief devices used to protect against overpressure in pressure vessels constructed to the requirements of this Division. It establishes the type, quantity, and settings of acceptable devices and the relieving capacity requirements for the applicable pressure vessels. Also provided are the requirements for capacity certification testing, as well as for obtaining and using the Code symbol stamp for pressure relief devices. In addition, this Part provides the requirements for installation of these pressure relief devices.

b) Unless otherwise defined in this Division, the definitions relating to pressure relief devices in Section 2 of ASME PTC 25 shall apply.

9.1.1 Protection Against Overpressure

a) All pressure vessels within the scope of this Division, irrespective of size or pressure, shall be provided with protection against overpressure in accordance with the requirements of this Part.

b) The vessel Manufacturer need not supply pressure relief devices or other overpressure protection. It is the responsibility of the user to ensure that the required pressure relief devices and/or overpressure protection are properly installed and in place prior to initial operation.

c) Pressure relief devices for vessels that are to operate completely filled with liquid shall be designed for liquid service, unless the vessel is otherwise protected against overpressure.

d) The protective devices provided in accordance with paragraph 9.1.1.a) need not be installed directly on a pressure vessel when the source of pressure is external to the vessel and is under such positive control that the pressure in the vessel cannot exceed the maximum allowable working pressure (MAWP) at the operating temperature except as permitted in Section VIII, Division 1. Note that pressure reducing valves and similar mechanical or electrical control instruments, except for pilot operated pressure relief valves, are not considered as sufficiently positive in action to prevent excess pressures from being developed.

e) Pressure relieving devices shall be constructed, located, and installed so that they are readily accessible for testing, inspection, replacement, and repair and so that they cannot be readily rendered inoperative (see Annex 9.A for the use of stop valves), and should be selected on the basis of their intended service.

f) It is the responsibility of the user or his/her designated agent to size and select the pressure relief device(s) or overpressure protection provisions based on its intended service. Intended service considerations shall include, but not necessarily be limited to the following:

   1) Normal operating and upset conditions

   2) Fluids

   3) Fluid phases

9.1.2 Types of Overpressure Protection

a) All pressure relief devices listed in Section VIII, Division 1 and bearing either the "UV" or "UD" Code Symbol Stamp are permissible.

b) Pressure relief valves certified for a steam discharging capacity under the provisions of Section I, and bearing the official Code symbol stamp of Section I for safety valves, may be used on pressure vessels constructed to this Division. The rated capacity in terms of other fluids shall be determined by the method of conversion given in Section VIII, Division 1, Appendix 11.

c) Where overpressure protection is provided by means other than the use of pressure relief devices, the requirements of paragraph 9.7 shall be followed.

9.1.3 Required Relieving Capacity and Allowable Overpressure

a) Relieving capacity and allowable overpressure shall be in accordance with the requirements specified in Section VIII, Division 1.

b) Where overpressure protection is provided by means other than the use of pressure relief devices, the requirements of paragraph 9.7 shall be followed and the allowable overpressure (accumulation) shall not exceed the maximum allowable working pressure.

9.1.4 Pressure Setting of Pressure Relief Devices

All pressure relief devices shall follow all requirements of Section VIII, Division 1 for pressure setting including tolerances.

9.2 Pressure Relief Valves

Except as permitted by paragraph 9.1.2.b), safety, safety relief, relief and pilot-operated pressure relief valves shall be as defined in Section VIII, Division 1, and shall meet all requirements of Section VIII, Division 1.

9.3 Non-Reclosing Pressure Relief Devices

9.3.1 Rupture Disk Devices

Rupture disk devices and rupture disk holders shall be as defined in Section VIII, Division 1, and shall meet all requirements for application, burst pressure, certification and installation of Section VIII, Division 1.

9.3.2 Breaking Pin Devices

9.3.2.1 General

Breaking pin devices and breaking pin housings shall be as defined in Section VIII, Division 1, and shall meet all requirements for application, break pressure and certification of flow capacity of Section VIII, Division 1.

9.3.2.2 Determination of Rated Flow Capacity

The capacity of a pressure relief valve and breaking pin combination based on an in-service fluid or in-service conditions different from those of the certification tests shall be calculated using the conversion methods provided in Section VIII, Division 1, Appendix 11.

9.3.3 Spring Loaded Non-Reclosing Pressure Relief Devices

Spring loaded non-reclosing pressure relief devices shall be as defined in Section VIII, Division 1, and shall meet all requirements for application, set pressure, capacity certification and tolerance of Section VIII, Division 1.

9.4 Calculation Of Rated Capacity For Different Relieving Pressures And/Or Fluids

9.4.1 General

Determination of rated capacity of a pressure relief device at relieving pressures other than 110% of set pressure shall be performed in accordance with the requirements of Section VIII, Division 1.

9.4.2 Prorating of Certified Capacity for Different Pressures

Determination of the relieving capacity of a pressure relief device for in-service fluids other than steam or air shall be determined by the conversion method of Section VIII, Division 1, Appendix 11.

9.4.3 Conversion of Certified Capacity for Different In-Service Fluids

The relieving capacity of a pressure relief device for in-service fluids other than steam or air shall be determined by the method of conversion given in Section VIII, Division 1, Appendix 11.

9.5 Marking and Stamping

Except as permitted by paragraph 9.1.2.b), all pressure relief devices used shall be marked and stamped in accordance with the requirements of Section VIII, Division 1.

9.6 Provisions for Installation of Pressure Relieving Devices

9.6.1 General

Pressure relief device Installation shall comply with Section VIII, Division 1.

9.6.2 Inlet Piping for Pressure Relief Devices

The design of inlet piping for pressure relief devices shall be in accordance with the requirements of Section VIII, Division 1. Additional guidance is provided in Annex 9.A.

9.6.3 Discharge Lines from Pressure Relief Devices

The design of discharge piping from pressure relief devices shall be in accordance with the requirements of Section VIII, Division 1. Additional guidance is provided in Annex 9.A.

9.6.4 Pressure Drop, Non-Reclosing Pressure Relief Devices

Piping, valves and fittings, and vessel components comprising part of a non-reclosing device pressure relieving system shall be sized to prevent the vessel pressure from rising above the allowable overpressure.

9.7 Overpressure Protection by Design

A pressure vessel may be provided with overpressure protection by system design in lieu of a pressure relief device or pressure relief devices if all provisions of paragraph UG-140 of Section VIII Division 1 are satisfied.

ANNEX 9.A

BEST PRACTICES FOR THE INSTALLATION AND OPERATION OF PRESSURE RELIEF DEVICES

(INFORMATIVE)

9.A.1 Introduction

This Annex provides additional guidance for design of pressure relief device installations. This Annex is a supplement to the installation requirements provided in Part 9. Note that there may be jurisdictional requirements related to the installation of pressure relief devices.

9.A.2 Provisions for the Installation of Stop Valves in the Relief Path

9.A.2.1 General

The general provisions for the installation of pressure relieving devices are covered in paragraph 9.6. The following paragraphs contain requirements for system and stop valve design when stop valves are to be located within the relief path. These stop valves are sometimes necessary for the continuous operation of processing equipment of such a complex nature that shutdown of any part of it is not feasible or not practical. The requirements cover stop valves provided upstream and downstream of pressure relief valves, provided in the relief path where there is normally a process flow and in a relief path where fire is the only potential source of overpressure.

9.A.2.2 Stop Valves Located in the Relief Path

9.A.2.2.1 General

a) A stop valve(s) located within the relief path is not allowed except as permitted by paragraphs 9.A.2.2.5, 9.A.2.2.6, 9.A.2.2.7 and 9.A.2.2.8 below, and only when specified by the user. The responsibilites of the user are summarized in paragraph 9.A.2.2.3. The specific requirements of paragraphs 9.A.2.2.5, 9.A.2.2.6, 9.A.2.2.7 and 9.A.2.2.8 are not intended to allow for operation above the maximum allowable working pressure.

b) The pressure relief path shall be designed such that the pressure in the equipment being protected does not exceed the maximum allowable working pressure before the pressure at the pressure relief device reaches its set pressure and the pressure does not exceed the allowable overpressure limits of Section VIII, Division 1.

9.A.2.2.2 Definitions

a) *Administrative Controls* are procedures that, in combination with mechanical locking elements, are intended to ensure that personnel actions do not compromise the overpressure protection of the equipment. They include, as a minimum:

   1) Documented Operation and Maintenance Procedures,

   2) Operator and Maintenance Personnel Training in the above procedures.

b) The *Pressure Relief Path* consists of all equipment, pipe, fittings and valves in the flow path between any protected equipment item and its pressure relieving device and the pressure relieving device and the discharge point of the relieving stream. Stop valves within a pressure relief path include, but are not limited to, those located directly upstream and downstream of the pressure relief device that may be provided exclusively for pressure relief device maintenance.

c) *Valve Operation Controls* are devices used to ensure that stop valves within the pressure relief path are in their proper (open/closed) position. They include the following:

   1) Mechanical Interlocks which are designed to prevent valve operations which could result in the blocking of a pressure relief path before an alternative pressure relief path is put into service.

   2) Instrumented Interlocks which function similar to mechanical interlocks, except that instrument permissives and/or overrides are used instead of mechanical linkages/devices to prevent valve positions that block the pressure relief path.

   3) Three-way valves designed to prevent a flow path from being blocked without another flow path being simultaneously opened.

d) *Valve Failure Controls* are measures taken in valve design, configuration, and/or orientation of the stop valve with the purpose of preventing an internal failure of a stop valve from closing and blocking the pressure relief path. An example of valve failure controls is the installation of gate valves with the stem oriented at or below the horizontal position.

e) A *Full Area Stop Valve* is a valve in which the flow area of the valve is equal to or larger than the inlet flow area of the pressure relief device.

f) *Mechanical Locking Elements* are elements that when installed on a stop valve, provide a physical barrier to the operation of the stop valve, such that the stop valve is not capable of being operated unless a deliberate action is taken to remove or disable the element. Such elements when used in combination with administrative controls, ensure that the equipment overpressure protection is not compromised by personnel actions. Examples of mechanical locking elements include locks (with or without chains) on the stop valve handwheels, levers, or actuators, and plastic or metal straps (car seals) that are secured to the valve in such a way that the strap must be broken to operate the stop valve.

g) A *Management System* is the collective application of administrative controls, valve operation controls, and valve failure controls, in accordance with the applicable requirements of this Division.

9.A.2.2.3 Responsibilities

The user has the responsibility to establish and maintain a management system that ensures that a vessel is not operated without overpressure protection. These responsibilities include, but are not limited to, the following:

a) Deciding and specifying if the overpressure protection system will allow the use of stop valves(s) located in the relief path.

b) Establishing the pressure relief philosophy and the administrative controls requirements.

c) Establishing the required level of reliability, redundance, and maintenance of instrumented interlocks, if used.

d) Establishing procedures to ensure that the equipment is adequately protected against overpressure.

e) Ensuring that authorization to operate indentified valves is clear and that personnel are adequately trained for the task.

f) Establishing management systems to ensure that administrative controls are effective.

g) Establishing the analysis procedures and basis to be used in determining the potential levels of pressure if the stop valve(s) is closed.

h) Ensuring that the analysis described in paragraph 9.A.2.2.3.g is conducted by personnel who are qualified and experienced with the analysis procedure.

i) Ensuring that the other system components are acceptable for the potential levels of pressure established in paragraph 9.A.2.2.3.g.

j) Ensuring that the results of the analysis described in paragraph 9.A.2.2.3.g are documented and reviewed and accepted in writing by the individual responsible for the operation of the vessel and valves.

k) Ensuring that the administrative controls are reviewed and accepted in writing by the individual responsible for operation of the vessel and valves.

9.A.2.2.4 Requirements of Procedures/Management Systems

a) Procedures shall specify that valves requiring mechanical locking elements and/or valve operation controls and/or valve failure controls shall be documented and clearly identified as such.

b) The Management System shall document the administrative controls (training and procedures), the valve controls, and the performance of the administrative controls in an auditable form for management review.

9.A.2.2.5 Stop Valves Provided in Systems for Which the Pressure Originates Exclusively From an Outside Source

A vessel or system for which the pressure originates from an outside source exclusively may have individual pressure relieving devices on each vessel, or connected to any point on the connecting piping, or on any one of the vessels to be protected. Under such an arrangement, there may be stop valve(s) between any vessel and the pressure relieving devices, and these stop valve(s) need not have any administrative controls, valve operation controls, or valve failure controls, provided that the stop valves also isolate the vessel from the source of pressure.

9.A.2.2.6 Stop Valves Provided Upstream or Downstream of the Pressure Relief Device Exclusively for Maintenance of That Device

Full area stop valve(s) may be provided upstream and/or downstream of the pressure relieving device for the purpose of inspection, testing and repair of the pressure relief device or discharge header isolation, provided that, as a minimum, the following requirements are complied with:

c) Administrative controls are provided to prevent unauthorized valve operation.

d) Valves are provided with mechanical locking elements.

e) Valve failure controls are provided to prevent accidental valve closure due to mechanical failure.

f) Procedures are in place to provide pressure relief protection during the time when the system is isolated from its pressure relief path. These procedures shall ensure that when the system is isolated from its pressure relief path, an authorized person shall continuously monitor the pressure conditions of the vessel and shall be capable of responding promptly with documented, pre-defined actions, either stopping the source of overpressure or opening alternative means of pressure relief. This authorized person shall be dedicated to this task and shall have no other duties when performing this task.

g) The system shall be isolated from its pressure relief path for only the time required to test, repair, and/or replace the pressure relief device.

9.A.2.2.7 Stop Valves Provided in the Pressure Relief Path Where There is Normally Process Flow

Stop valve(s), excluding remotely operated valves, may be provided in the relief path where there is normally process flow, provided the requirements in paragraphs 9.A.2.2.7.a and 9.A.2.2.7.b, as a minimum, are complied with. These requirements are based on the overpressure scenarios involving accidental closure of a single stop valve within the relief path (see paragraph 9.A.2.2.3.g) The accidental closure of these stop valve(s) in the pressure relief system need not be considered in the determination of the specified design pressure in Part 2 of this Division.

a) The flow resistance of the valve in the full open position does not reduce the relieving capacity below that required by paragraph 9.1.3.

b) The closure of the valve will be readily apparent to the operators such that corrective action, in accordance with documented operating procedures, is required and:

    1) If the pressure due to closure of the valve cannot exceed 116% of the maximum allowable working pressure, then no administrative controls, or valve failure controls are required, or

2) If the pressure due to closure of the valve cannot exceed the following:

   i) the documented test pressure, multiplied by the ratio of stress value at the design temperature to the stress value at the test temperature, or

   ii) if the test pressure is calculated per Part 8, paragraph 8.2.1.e , in addition to the stress ratio specified in paragraph 9.A.2.2.7.b.2.i, the test pressure shall also be multiplied by the ratio of the nominal thickness minus the corrosion allowance to the nominal thickness then, as a minimum, administrative controls and mechanical locking elements are required, or

3) If the pressure due to closure of the valve could exceed the pressure in paragraph 9.A.2.2.7.b.2 , then the user shall either:

   i) eliminate the stop valve, or

   ii) apply administrative controls, mechanical locking elements, valve failure controls, and valve operation controls, or

   iii) provide a pressure relief device to protect the equipment that could be overpressured due to closure of the stop valve.

9.A.2.2.8 Stop Valves Provided in the Relief Path of Equipment Where Fire is the Only Potential Source of Overpressure

Full area stop valves located in the relief path of equipment where fire is the only potential source of overpressure do not require mechanical locking elements, valve operation controls, or valve failure controls provided the user has documented operating procedures requiring the equipment isolated from its pressure relief path is depressured and free of all liquids.

9.A.3 Inlet Piping Pressure Drop for Pressure Relief Valves

For pressure relief valves, the flow characteristics of the upstream system shall be such that the cumulative total of all non-recoverable inlet losses shall not exceed 3% of the valve set pressure. The inlet pressure losses shall be determined accounting for all fittings in the upstream system, including rupture disks installed in the pressure relief valve inlet piping, and shall be based on the valve nameplate capacity corrected for the characteristics of the flowing fluid.

9.A.4 Discharge Lines from Pressure Relief Devices

a) Where it is feasible, the use of a short discharge pipe or vertical riser, connected through long-radius elbows from each individual device, blowing directly to the atmosphere, is recommended. For pressure relief valves, such discharge pipes shall be at least of the same size as the valve outlet. Where the nature of the discharge permits, telescopic (sometimes called "broken") discharge lines, whereby condensed vapor in the discharge line, or rain, is collected in a drip pan and piped to a drain, are recommended. This construction has the further advantage of not transmitting discharge pipe strains to the pressure relief device. In these types of installations, the backpressure effect will be negligible, and no undue influence upon normal operation of the pressure relief device can result.

b) When discharge lines are long, or where outlets of two or more pressure relief devices are connected into a common line, the effect of the back pressure on pressure relief device operation and capacity shall be considered. The sizing of any section of a common discharge header downstream from each of the two or more pressure relief devices that may reasonably be expected to discharge simultaneously shall be based on the total of their outlet areas, with due allowance for the pressure drop in all downstream sections. Use of specially designed devices suitable for use on high or variable backpressure service should be considered.

c) The flow characteristics of the discharge system of high lift, top guided direct spring loaded pressure relief valves or pilot-operated pressure relief valves in compressible fluid service shall be such that the static pressure developed at the discharge flange of a conventional direct spring loaded pressure relief valve

will not exceed 10% of the set pressure when flowing at rated capacity. Other valve types exhibit various degrees of tolerance to back pressure and the Manufacturer's recommendation should be followed.

d) All discharge lines shall be run as directly as practicable to the point of final release for disposal. For the longer lines, due consideration shall be given to the advantage of long-radius elbows, avoidance of close-up fittings, minimizing line strains and using well-known means of support to minimize line sway and vibration under operating conditions.

e) Provisions should be made in all cases for adequate drainage of discharge lines.

f) It is recognized that no simple rule can be applied generally to fit the many installation requirements. Installations vary from simple short lines that discharge directly to the atmosphere to the extensive manifold discharge piping systems where the quantity and rate of the product to be disposed of requires piping to a distant safe place.

9.A.5 Cautions Regarding Pressure Relief Device Discharge into a Common Header

Because of the wide variety of types and kinds of pressure relief devices, it is not considered advisable to attempt a description of the effects produced by discharging them into a common header. Several different types of pressure relief devices may conceivably be connected into the same discharge header and the effect of backpressure on each type may be radically different. Data compiled by the Manufacturers of each type of pressure relief device used should be consulted for information relative to its performance under the conditions anticipated.

9.A.6 Pressure Differentials (Operating Margin) for Pressure Relief Valves

9.A.6.1 General

a) Due to the variety of service conditions and the various designs of pressure relief valves, only general guidance can be given regarding the differential between the set pressure of the pressure relief valve and the operating pressure of the vessel.

b) Providing an adequate pressure differential for the application will minimize operating difficulty. The following is general advisory information on the characteristics of the intended service and of the pressure relief valves that may bear on the proper pressure differential selection for a given application. These considerations should be reviewed early in the system design since they may dictate the maximum allowable working pressure of the system.

9.A.6.2 Considerations for Establishing the Operating Margin

9.A.6.2.1 Process Conditions

a) To minimize operational problems, the user should consider not only normal operating conditions of fluids, pressures, and temperatures, but also start-up and shutdown conditions, process upsets, anticipated ambient conditions, instrument response times, pressure surges due to quick closing valves, etc.

b) When such conditions are not considered, the pressure relief valve may become, in effect, a pressure controller, a duty for which it is not designed.

c) Additional consideration should be given to hazard and pollution associated with the release of the fluid. Larger differentials may be appropriate for fluids that are toxic, corrosive, or exceptionally valuable.

9.A.6.2.2 Pressure Relief Valve Characteristics

a) The blowdown characteristic and capability is the first consideration in selecting a compatible pressure relief valve and operating margin. After a self-actuated release of pressure, the pressure relief valve must be capable of reclosing above the normal operating pressure. For example, if the pressure relief valve is set at 690 kPa (100 psig) with a 7% blowdown, it will close at 641 kPa (93 psig). The operating

pressure must be maintained below 641 kPa (93 psig) in order to prevent leakage or flow from a partially open valve.

b) Users should exercise caution regarding the blowdown adjustment of large spring-loaded valves. Test facilities, whether owned by Manufacturers, repair houses, or users, may not have sufficient capacity to accurately verify the blowdown setting. The settings cannot be considered accurate unless made in the field on the actual installation.

c) Pilot-operated valves represent a special case from the standpoints of both blowdown and tightness. The pilot portion of some pilot-operated valves can be set at blowdowns as short as 2%. This characteristic is not, however, reflected in the operation of the main valve in all cases. The main valve can vary considerably from the pilot depending on the location of the two components in the system. If the pilot is installed remotely from the main valve, significant time and pressure lags can occur, but reseating of the pilot assures reseating of the main valve. The pressure drop in the connecting piping between the pilot and the main valve must not be excessive; otherwise, the operation of the main valve will be adversely affected. The tightness of the main valve portion of these combinations is considerably improved above that of conventional valves by pressure loading the main disk or by the use of soft seats or both. Despite the apparent advantages of pilot-operated valves, users should be aware that they should not be employed in abrasive or dirty service, in applications where coking, polymerization, or corrosion of the wetted pilot parts can occur, or where freezing or condensation of the fluid at ambient temperatures is possible. For all applications, the pressure relief valve Manufacturer should be consulted prior to selecting a valve of this type.

d) Tightness capability is another factor affecting valve selection, whether spring-loaded or pilot-operated. It varies somewhat depending on whether metal or resilient seats are specified, and also on such factors as corrosion or temperature. The required tightness and test method should be specified to comply at a pressure no lower than the normal operating pressure of the process. A recommended procedure and acceptance standard is given in API Standard 527, Seat Tightness of Pressure Relief Valves. It should also be noted that any degree of tightness obtained should not be considered permanent. Service operation of a valve almost invariably reduces the degree of tightness.

e) Application of special designs such as O-rings or resilient seats should be reviewed with the pressure relief valve Manufacturer.

f) The anticipated behavior of the pressure relief valves includes allowance for a plus-or-minus tolerance on set pressure that varies with the pressure level. Installation conditions, such as backpressure, variations, and vibrations influence selection of special designs and may require an increase in the differential pressure (operating margin).

9.A.6.2.3 General Recommendations for Pressure Differentials (Operating Margin)

The following pressure differentials are recommended unless the pressure relief valve has been designed or tested in a specific or similar service, and a smaller differential has been recommended by the Manufacturer.

a) A minimum difference of 35 kPa (5 psi) is recommended for set pressures to 485 kPa (70 psi). In this category, the set pressure tolerance is $\pm$ 13.8 kPa ($\pm$2 psi), and the differential to the leak test pressure is 10% or 35 kPa (5 psi), whichever is greater.

b) A minimum differential of 10% is recommended for set pressures from 490 to 6900 kPa (71 psi to 1000 psi). In this category, the set pressure tolerance is $\pm$ 3% and the differential to the leak test pressure is 10%.

c) A minimum differential of 7% is recommended for set pressures above 6900 kPa (1000 psi). In this category, the set pressure tolerance is $\pm$ 3% and the differential to the leak test pressure is 5%.

d) Pressure relief valves having small seat sizes will require additional maintenance when the pressure differential approaches these recommendations.

9.A.7 Pressure Relief Valve Orientation

Spring-loaded pressure relief valves normally should be installed in the upright position with the spindle vertical. Where space or piping configuration preclude such an installation, the valve may be installed in other than the vertical position provided that:

a) The pressure relief valve design is satisfactory for such position and is acceptable to the Manufacturer of the valve,

b) The media is such that solid material will not accumulate at the inlet of the pressure relief valve, and

c) Drainage of the discharge side of the pressure relief valve body and discharge piping prevents collection of liquid on the valve disk or in the discharge piping.

9.A.8 Reaction Forces and Externally Applied Piping Loads

a) The discharge of a pressure relief device imposes reactive flow forces on the device and associated piping. The design of the installation may require computation of the bending moments and stresses in the piping and vessel nozzle. There are momentum effects and pressure effects at steady state flow as well as transient dynamic loads caused by opening.

b) Mechanical forces may be applied to the pressure relief device by discharge piping as a result of thermal expansion, movement away from anchors, and weight of any unsupported piping. The resultant bending moments on a closed pressure relief device may cause leakage, device damage, and excessive stress in inlet piping. The design of the installation should consider these possibilities.

9.A.9 Sizing of Pressure Relief Devices for Fire Conditions

a) Excessive pressure may develop in pressure vessels by vaporization of the liquid contents and/or expansion of vapor content due to heat influx from the surroundings, particularly from a fire.

b) Pressure relief systems for fire conditions are usually intended to release only the quantity of product necessary to lower the pressure to a predetermined safe level, without releasing an excessive quantity. This control is especially important in situations where release of the contents generates a hazard because of flammability or toxicity.

c) Under fire conditions, consideration must also be given to the possibility that the safe pressure level for the vessel will be reduced due to heating of the vessel material, with a corresponding loss of strength.

d) Several equations have evolved over the years for calculating the pressure relief capacity required under fire conditions. The major differences involve heat flux rates. There is no single equation yet developed which takes into account all of the many factors that could be considered in making this determination. When fire conditions are a consideration in the design of a pressure vessel, the following references which provide recommendations for specific installations may be used:

    1) API Recommended Practice 520, Sizing, Selection, and Installation of Pressure-Relieving Systems in Refineries, Part 1 – Sizing and Selection, Seventh Edition, January 2000, American Petroleum Institute, Washington, DC.

    2) API Standard 521, Pressure-Relieving and Depressuring Systems, Fifth Edition, Jan. 2007, American Petroleum Institute, Washington, DC.

    3) API Standard 2000, Venting Atmospheric and Low-Pressure Tanks (Nonrefrigerated and Refrigerated), Fifth edition, April 1998, American Petroleum Institute, Washington, DC

    4) AAR Standard M-1002, Specifications for Tank Cars, 1978, Association of American Railroads, Washington, DC.

5) Safety Relief Device Standards: S-l.l, Cylinders for Compressed Gases; S-1.2, Cargo and Portable Tanks; and S-1.3, Compressed Gas Storage Containers, Compressed Gas Association, Arlington, VA.

6) NFPA Code Nos. 30, 58, 59, and 59A, National Fire Protection Association, 1 Batterymarch Park, Quincy, MA, 02169-7471.

7) *Pressure-Relieving Systems for Marine Cargo Bulk Liquid Containers,* 1973, National Academy of Sciences, Washington, DC.

9.A.10 Use of Pressure Indicating Devices to Monitor Pressure Differential

If a pressure indicating device is provided to monitor the vessel pressure at or near the set pressure of the pressure relief device, one should be selected that spans the set pressure of the pressure relief device and is graduated with an upper limit that is neither less than 1.25 times the set pressure of the pressure relief device nor more than twice the maximum allowable working pressure of the vessel. Additional devices may be installed if desired.

ANNEX 9.A

BEST PRACTICES FOR THE INSTALLATION AND OPERATION OF PRESSURE RELIEF DEVICES

(INFORMATIVE)

9.A.1 Introduction

This Annex provides additional guidance for design of pressure relief device installations. This Annex is a supplement to the installation requirements provided in Part 9. Note that there may be jurisdictional requirements related to the installation of pressure relief devices.

9.A.2 Provisions for the Installation of Stop Valves in the Relief Path

9.A.2.1 General

The general provisions for the installation of pressure relieving devices are covered in paragraph 9.6. The following paragraphs contain requirements for system and stop valve design when stop valves are to be located within the relief path. These stop valves are sometimes necessary for the continuous operation of processing equipment of such a complex nature that shutdown of any part of it is not feasible or not practical. The requirements cover stop valves provided upstream and downstream of pressure relief valves, provided in the relief path where there is normally a process flow and in a relief path where fire is the only potential source of overpressure.

9.A.2.2 Stop Valves Located in the Relief Path

9.A.2.2.1 General

a) A stop valve(s) located within the relief path is not allowed except as permitted by paragraphs 9.A.2.2.5, 9.A.2.2.6, 9.A.2.2.7 and 9.A.2.2.8 below, and only when specified by the user. The responsibilites of the user are summarized in paragraph 9.A.2.2.3. The specific requirements of paragraphs 9.A.2.2.5, 9.A.2.2.6, 9.A.2.2.7 and 9.A.2.2.8 are not intended to allow for operation above the maximum allowable working pressure.

b) The pressure relief path shall be designed such that the pressure in the equipment being protected does not exceed the maximum allowable working pressure before the pressure at the pressure relief device reaches its set pressure and the pressure does not exceed the allowable overpressure limits of Section VIII, Division 1.

9.A.2.2.2 Definitions

a) *Administrative Controls* are procedures that, in combination with mechanical locking elements, are intended to ensure that personnel actions do not compromise the overpressure protection of the equipment. They include, as a minimum:

   1) Documented Operation and Maintenance Procedures,

   2) Operator and Maintenance Personnel Training in the above procedures.

b) The *Pressure Relief Path* consists of all equipment, pipe, fittings and valves in the flow path between any protected equipment item and its pressure relieving device and the pressure relieving device and the discharge point of the relieving stream. Stop valves within a pressure relief path include, but are not limited to, those located directly upstream and downstream of the pressure relief device that may be provided exclusively for pressure relief device maintenance.

c) *Valve Operation Controls* are devices used to ensure that stop valves within the pressure relief path are in their proper (open/closed) position. They include the following:

   1) Mechanical Interlocks which are designed to prevent valve operations which could result in the blocking of a pressure relief path before an alternative pressure relief path is put into service.

   2) Instrumented Interlocks which function similar to mechanical interlocks, except that instrument permissives and/or overrides are used instead of mechanical linkages/devices to prevent valve positions that block the pressure relief path.

   3) Three-way valves designed to prevent a flow path from being blocked without another flow path being simultaneously opened.

d) *Valve Failure Controls* are measures taken in valve design, configuration, and/or orientation of the stop valve with the purpose of preventing an internal failure of a stop valve from closing and blocking the pressure relief path. An example of valve failure controls is the installation of gate valves with the stem oriented at or below the horizontal position.

e) A *Full Area Stop Valve* is a valve in which the flow area of the valve is equal to or larger than the inlet flow area of the pressure relief device.

f) *Mechanical Locking Elements* are elements that when installed on a stop valve, provide a physical barrier to the operation of the stop valve, such that the stop valve is not capable of being operated unless a deliberate action is taken to remove or disable the element. Such elements when used in combination with administrative controls, ensure that the equipment overpressure protection is not compromised by personnel actions. Examples of mechanical locking elements include locks (with or without chains) on the stop valve handwheels, levers, or actuators, and plastic or metal straps (car seals) that are secured to the valve in such a way that the strap must be broken to operate the stop valve.

g) A *Management System* is the collective application of administrative controls, valve operation controls, and valve failure controls, in accordance with the applicable requirements of this Division.

9.A.2.2.3 Responsibilities

The user has the responsibility to establish and maintain a management system that ensures that a vessel is not operated without overpressure protection. These responsibilities include, but are not limited to, the following:

a) Deciding and specifying if the overpressure protection system will allow the use of stop valves(s) located in the relief path.

b) Establishing the pressure relief philosophy and the administrative controls requirements.

c) Establishing the required level of reliability, redundance, and maintenance of instrumented interlocks, if used.

d) Establishing procedures to ensure that the equipment is adequately protected against overpressure.

e) Ensuring that authorization to operate indentified valves is clear and that personnel are adequately trained for the task.

f) Establishing management systems to ensure that administrative controls are effective.

g) Establishing the analysis procedures and basis to be used in determining the potential levels of pressure if the stop valve(s) is closed.

h) Ensuring that the analysis described in paragraph 9.A.2.2.3.g is conducted by personnel who are qualified and experienced with the analysis procedure.

i) Ensuring that the other system components are acceptable for the potential levels of pressure established in paragraph 9.A.2.2.3.g.

j) Ensuring that the results of the analysis described in paragraph 9.A.2.2.3.g are documented and reviewed and accepted in writing by the individual responsible for the operation of the vessel and valves.

k) Ensuring that the administrative controls are reviewed and accepted in writing by the individual responsible for operation of the vessel and valves.

9.A.2.2.4 Requirements of Procedures/Management Systems

a) Procedures shall specify that valves requiring mechanical locking elements and/or valve operation controls and/or valve failure controls shall be documented and clearly identified as such.

b) The Management System shall document the administrative controls (training and procedures), the valve controls, and the performance of the administrative controls in an auditable form for management review.

9.A.2.2.5 Stop Valves Provided in Systems for Which the Pressure Originates Exclusively From an Outside Source

A vessel or system for which the pressure originates from an outside source exclusively may have individual pressure relieving devices on each vessel, or connected to any point on the connecting piping, or on any one of the vessels to be protected. Under such an arrangement, there may be stop valve(s) between any vessel and the pressure relieving devices, and these stop valve(s) need not have any administrative controls, valve operation controls, or valve failure controls, provided that the stop valves also isolate the vessel from the source of pressure.

9.A.2.2.6 Stop Valves Provided Upstream or Downstream of the Pressure Relief Device Exclusively for Maintenance of That Device

Full area stop valve(s) may be provided upstream and/or downstream of the pressure relieving device for the purpose of inspection, testing and repair of the pressure relief device or discharge header isolation, provided that, as a minimum, the following requirements are complied with:

c) Administrative controls are provided to prevent unauthorized valve operation.

d) Valves are provided with mechanical locking elements.

e) Valve failure controls are provided to prevent accidental valve closure due to mechanical failure.

f) Procedures are in place to provide pressure relief protection during the time when the system is isolated from its pressure relief path. These procedures shall ensure that when the system is isolated from its pressure relief path, an authorized person shall continuously monitor the pressure conditions of the vessel and shall be capable of responding promptly with documented, pre-defined actions, either stopping the source of overpressure or opening alternative means of pressure relief. This authorized person shall be dedicated to this task and shall have no other duties when performing this task.

g) The system shall be isolated from its pressure relief path for only the time required to test, repair, and/or replace the pressure relief device.

9.A.2.2.7 Stop Valves Provided in the Pressure Relief Path Where There is Normally Process Flow

Stop valve(s), excluding remotely operated valves, may be provided in the relief path where there is normally process flow, provided the requirements in paragraphs 9.A.2.2.7.a and 9.A.2.2.7.b, as a minimum, are complied with. These requirements are based on the overpressure scenarios involving accidental closure of a single stop valve within the relief path (see paragraph 9.A.2.2.3.g) The accidental closure of these stop valve(s) in the pressure relief system need not be considered in the determination of the specified design pressure in Part 2 of this Division.

a) The flow resistance of the valve in the full open position does not reduce the relieving capacity below that required by paragraph 9.1.3.

b) The closure of the valve will be readily apparent to the operators such that corrective action, in accordance with documented operating procedures, is required and:

    1) If the pressure due to closure of the valve cannot exceed 116% of the maximum allowable working pressure, then no administrative controls, or valve failure controls are required, or

2) If the pressure due to closure of the valve cannot exceed the following:

   i) the documented test pressure, multiplied by the ratio of stress value at the design temperature to the stress value at the test temperature, or

   ii) if the test pressure is calculated per Part 8, paragraph 8.2.1.e , in addition to the stress ratio specified in paragraph 9.A.2.2.7.b.2.i, the test pressure shall also be multiplied by the ratio of the nominal thickness minus the corrosion allowance to the nominal thickness then, as a minimum, administrative controls and mechanical locking elements are required, or

3) If the pressure due to closure of the valve could exceed the pressure in paragraph 9.A.2.2.7.b.2 , then the user shall either:

   i) eliminate the stop valve, or

   ii) apply administrative controls, mechanical locking elements, valve failure controls, and valve operation controls, or

   iii) provide a pressure relief device to protect the equipment that could be overpressured due to closure of the stop valve.

9.A.2.2.8 Stop Valves Provided in the Relief Path of Equipment Where Fire is the Only Potential Source of Overpressure

Full area stop valves located in the relief path of equipment where fire is the only potential source of overpressure do not require mechanical locking elements, valve operation controls, or valve failure controls provided the user has documented operating procedures requiring the equipment isolated from its pressure relief path is depressured and free of all liquids.

9.A.3 Inlet Piping Pressure Drop for Pressure Relief Valves

For pressure relief valves, the flow characteristics of the upstream system shall be such that the cumulative total of all non-recoverable inlet losses shall not exceed 3% of the valve set pressure. The inlet pressure losses shall be determined accounting for all fittings in the upstream system, including rupture disks installed in the pressure relief valve inlet piping, and shall be based on the valve nameplate capacity corrected for the characteristics of the flowing fluid.

9.A.4 Discharge Lines from Pressure Relief Devices

a) Where it is feasible, the use of a short discharge pipe or vertical riser, connected through long-radius elbows from each individual device, blowing directly to the atmosphere, is recommended. For pressure relief valves, such discharge pipes shall be at least of the same size as the valve outlet. Where the nature of the discharge permits, telescopic (sometimes called "broken") discharge lines, whereby condensed vapor in the discharge line, or rain, is collected in a drip pan and piped to a drain, are recommended. This construction has the further advantage of not transmitting discharge pipe strains to the pressure relief device. In these types of installations, the backpressure effect will be negligible, and no undue influence upon normal operation of the pressure relief device can result.

b) When discharge lines are long, or where outlets of two or more pressure relief devices are connected into a common line, the effect of the back pressure on pressure relief device operation and capacity shall be considered. The sizing of any section of a common discharge header downstream from each of the two or more pressure relief devices that may reasonably be expected to discharge simultaneously shall be based on the total of their outlet areas, with due allowance for the pressure drop in all downstream sections. Use of specially designed devices suitable for use on high or variable backpressure service should be considered.

c) The flow characteristics of the discharge system of high lift, top guided direct spring loaded pressure relief valves or pilot-operated pressure relief valves in compressible fluid service shall be such that the static pressure developed at the discharge flange of a conventional direct spring loaded pressure relief valve

will not exceed 10% of the set pressure when flowing at rated capacity. Other valve types exhibit various degrees of tolerance to back pressure and the Manufacturer's recommendation should be followed.

d) All discharge lines shall be run as directly as practicable to the point of final release for disposal. For the longer lines, due consideration shall be given to the advantage of long-radius elbows, avoidance of close-up fittings, minimizing line strains and using well-known means of support to minimize line sway and vibration under operating conditions.

e) Provisions should be made in all cases for adequate drainage of discharge lines.

f) It is recognized that no simple rule can be applied generally to fit the many installation requirements. Installations vary from simple short lines that discharge directly to the atmosphere to the extensive manifold discharge piping systems where the quantity and rate of the product to be disposed of requires piping to a distant safe place.

9.A.5 Cautions Regarding Pressure Relief Device Discharge into a Common Header

Because of the wide variety of types and kinds of pressure relief devices, it is not considered advisable to attempt a description of the effects produced by discharging them into a common header. Several different types of pressure relief devices may conceivably be connected into the same discharge header and the effect of backpressure on each type may be radically different. Data compiled by the Manufacturers of each type of pressure relief device used should be consulted for information relative to its performance under the conditions anticipated.

9.A.6 Pressure Differentials (Operating Margin) for Pressure Relief Valves

9.A.6.1 General

a) Due to the variety of service conditions and the various designs of pressure relief valves, only general guidance can be given regarding the differential between the set pressure of the pressure relief valve and the operating pressure of the vessel.

b) Providing an adequate pressure differential for the application will minimize operating difficulty. The following is general advisory information on the characteristics of the intended service and of the pressure relief valves that may bear on the proper pressure differential selection for a given application. These considerations should be reviewed early in the system design since they may dictate the maximum allowable working pressure of the system.

9.A.6.2 Considerations for Establishing the Operating Margin

9.A.6.2.1 Process Conditions

a) To minimize operational problems, the user should consider not only normal operating conditions of fluids, pressures, and temperatures, but also start-up and shutdown conditions, process upsets, anticipated ambient conditions, instrument response times, pressure surges due to quick closing valves, etc.

b) When such conditions are not considered, the pressure relief valve may become, in effect, a pressure controller, a duty for which it is not designed.

c) Additional consideration should be given to hazard and pollution associated with the release of the fluid. Larger differentials may be appropriate for fluids that are toxic, corrosive, or exceptionally valuable.

9.A.6.2.2 Pressure Relief Valve Characteristics

a) The blowdown characteristic and capability is the first consideration in selecting a compatible pressure relief valve and operating margin. After a self-actuated release of pressure, the pressure relief valve must be capable of reclosing above the normal operating pressure. For example, if the pressure relief valve is set at 690 kPa (100 psig) with a 7% blowdown, it will close at 641 kPa (93 psig). The operating

pressure must be maintained below 641 kPa (93 psig) in order to prevent leakage or flow from a partially open valve.

b) Users should exercise caution regarding the blowdown adjustment of large spring-loaded valves. Test facilities, whether owned by Manufacturers, repair houses, or users, may not have sufficient capacity to accurately verify the blowdown setting. The settings cannot be considered accurate unless made in the field on the actual installation.

c) Pilot-operated valves represent a special case from the standpoints of both blowdown and tightness. The pilot portion of some pilot-operated valves can be set at blowdowns as short as 2%. This characteristic is not, however, reflected in the operation of the main valve in all cases. The main valve can vary considerably from the pilot depending on the location of the two components in the system. If the pilot is installed remotely from the main valve, significant time and pressure lags can occur, but reseating of the pilot assures reseating of the main valve. The pressure drop in the connecting piping between the pilot and the main valve must not be excessive; otherwise, the operation of the main valve will be adversely affected. The tightness of the main valve portion of these combinations is considerably improved above that of conventional valves by pressure loading the main disk or by the use of soft seats or both. Despite the apparent advantages of pilot-operated valves, users should be aware that they should not be employed in abrasive or dirty service, in applications where coking, polymerization, or corrosion of the wetted pilot parts can occur, or where freezing or condensation of the fluid at ambient temperatures is possible. For all applications, the pressure relief valve Manufacturer should be consulted prior to selecting a valve of this type.

d) Tightness capability is another factor affecting valve selection, whether spring-loaded or pilot-operated. It varies somewhat depending on whether metal or resilient seats are specified, and also on such factors as corrosion or temperature. The required tightness and test method should be specified to comply at a pressure no lower than the normal operating pressure of the process. A recommended procedure and acceptance standard is given in API Standard 527, Seat Tightness of Pressure Relief Valves. It should also be noted that any degree of tightness obtained should not be considered permanent. Service operation of a valve almost invariably reduces the degree of tightness.

e) Application of special designs such as O-rings or resilient seats should be reviewed with the pressure relief valve Manufacturer.

f) The anticipated behavior of the pressure relief valves includes allowance for a plus-or-minus tolerance on set pressure that varies with the pressure level. Installation conditions, such as backpressure, variations, and vibrations influence selection of special designs and may require an increase in the differential pressure (operating margin).

9.A.6.2.3 General Recommendations for Pressure Differentials (Operating Margin)

The following pressure differentials are recommended unless the pressure relief valve has been designed or tested in a specific or similar service, and a smaller differential has been recommended by the Manufacturer.

a) A minimum difference of 35 kPa (5 psi) is recommended for set pressures to 485 kPa (70 psi). In this category, the set pressure tolerance is $\pm$ 13.8 kPa ($\pm$2 psi), and the differential to the leak test pressure is 10% or 35 kPa (5 psi), whichever is greater.

b) A minimum differential of 10% is recommended for set pressures from 490 to 6900 kPa (71 psi to 1000 psi). In this category, the set pressure tolerance is $\pm$ 3% and the differential to the leak test pressure is 10%.

c) A minimum differential of 7% is recommended for set pressures above 6900 kPa (1000 psi). In this category, the set pressure tolerance is $\pm$ 3% and the differential to the leak test pressure is 5%.

d) Pressure relief valves having small seat sizes will require additional maintenance when the pressure differential approaches these recommendations.

9.A.7 Pressure Relief Valve Orientation

Spring-loaded pressure relief valves normally should be installed in the upright position with the spindle vertical. Where space or piping configuration preclude such an installation, the valve may be installed in other than the vertical position provided that:

a) The pressure relief valve design is satisfactory for such position and is acceptable to the Manufacturer of the valve,

b) The media is such that solid material will not accumulate at the inlet of the pressure relief valve, and

c) Drainage of the discharge side of the pressure relief valve body and discharge piping prevents collection of liquid on the valve disk or in the discharge piping.

9.A.8 Reaction Forces and Externally Applied Piping Loads

a) The discharge of a pressure relief device imposes reactive flow forces on the device and associated piping. The design of the installation may require computation of the bending moments and stresses in the piping and vessel nozzle. There are momentum effects and pressure effects at steady state flow as well as transient dynamic loads caused by opening.

b) Mechanical forces may be applied to the pressure relief device by discharge piping as a result of thermal expansion, movement away from anchors, and weight of any unsupported piping. The resultant bending moments on a closed pressure relief device may cause leakage, device damage, and excessive stress in inlet piping. The design of the installation should consider these possibilities.

9.A.9 Sizing of Pressure Relief Devices for Fire Conditions

a) Excessive pressure may develop in pressure vessels by vaporization of the liquid contents and/or expansion of vapor content due to heat influx from the surroundings, particularly from a fire.

b) Pressure relief systems for fire conditions are usually intended to release only the quantity of product necessary to lower the pressure to a predetermined safe level, without releasing an excessive quantity. This control is especially important in situations where release of the contents generates a hazard because of flammability or toxicity.

c) Under fire conditions, consideration must also be given to the possibility that the safe pressure level for the vessel will be reduced due to heating of the vessel material, with a corresponding loss of strength.

d) Several equations have evolved over the years for calculating the pressure relief capacity required under fire conditions. The major differences involve heat flux rates. There is no single equation yet developed which takes into account all of the many factors that could be considered in making this determination. When fire conditions are a consideration in the design of a pressure vessel, the following references which provide recommendations for specific installations may be used:

    1) API Recommended Practice 520, Sizing, Selection, and Installation of Pressure-Relieving Systems in Refineries, Part 1 – Sizing and Selection, Seventh Edition, January 2000, American Petroleum Institute, Washington, DC.

    2) API Standard 521, Pressure-Relieving and Depressuring Systems, Fifth Edition, Jan. 2007, American Petroleum Institute, Washington, DC.

    3) API Standard 2000, Venting Atmospheric and Low-Pressure Tanks (Nonrefrigerated and Refrigerated), Fifth edition, April 1998, American Petroleum Institute, Washington, DC

    4) AAR Standard M-1002, Specifications for Tank Cars, 1978, Association of American Railroads, Washington, DC.

5) Safety Relief Device Standards: S-l.l, Cylinders for Compressed Gases; S-1.2, Cargo and Portable Tanks; and S-1.3, Compressed Gas Storage Containers, Compressed Gas Association, Arlington, VA.

6) NFPA Code Nos. 30, 58, 59, and 59A, National Fire Protection Association, 1 Batterymarch Park, Quincy, MA, 02169-7471.

7) *Pressure-Relieving Systems for Marine Cargo Bulk Liquid Containers,* 1973, National Academy of Sciences, Washington, DC.

9.A.10 Use of Pressure Indicating Devices to Monitor Pressure Differential

If a pressure indicating device is provided to monitor the vessel pressure at or near the set pressure of the pressure relief device, one should be selected that spans the set pressure of the pressure relief device and is graduated with an upper limit that is neither less than 1.25 times the set pressure of the pressure relief device nor more than twice the maximum allowable working pressure of the vessel. Additional devices may be installed if desired.

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ISBN 978-0-7918-3252-3

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